AUG 9 1943

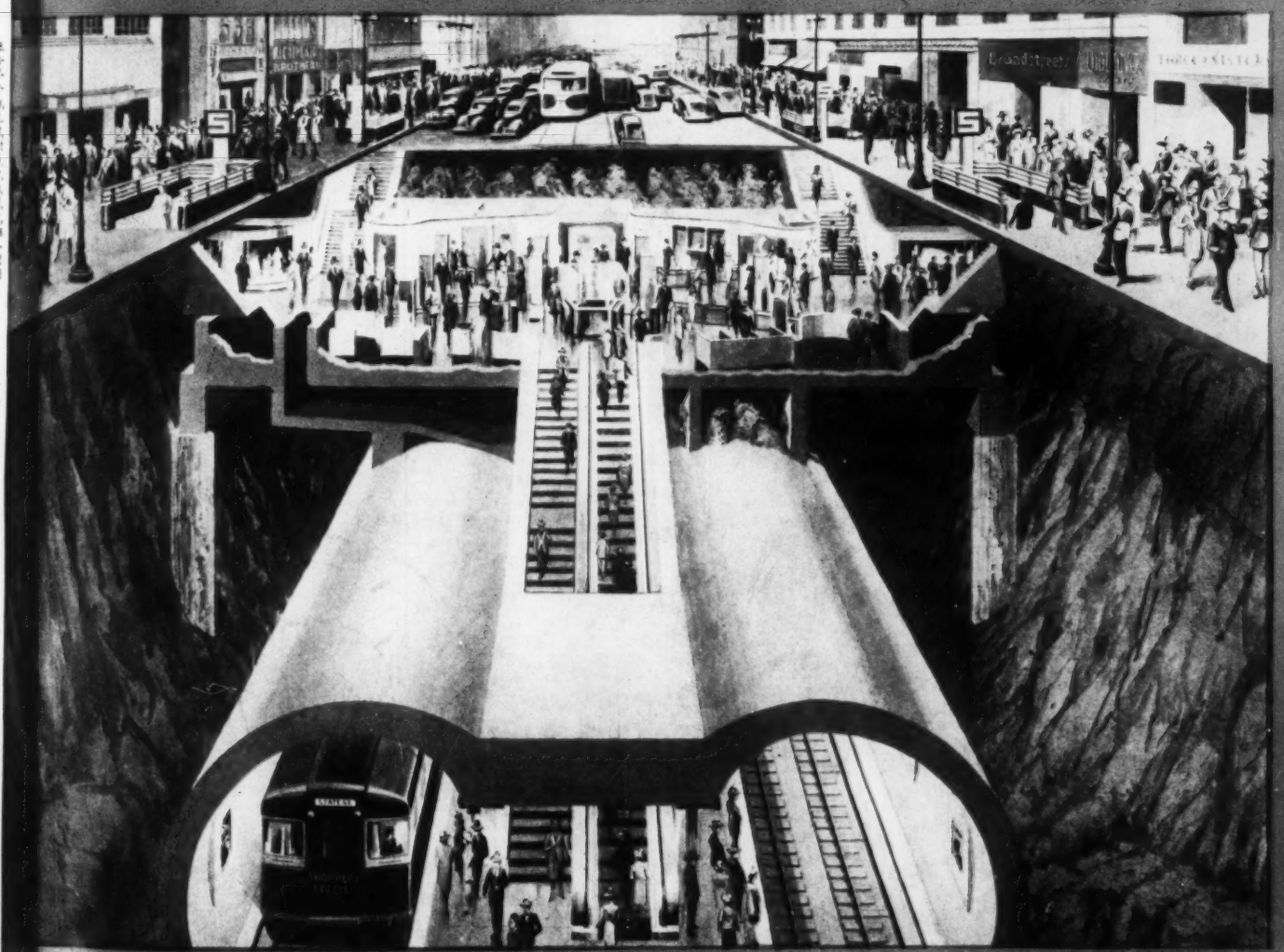
Heating Piping & Air Conditioning

DESIGN
INSTALLATION
OPERATION
MAINTENANCE

AUGUST ★ 1943

INDUSTRIAL
COMMERCIAL
AND LARGE
BUILDINGS

423: Journal Section of the American Society of Heating and Ventilating Engineers



A most interesting discussion of subway ventilation is among the many subjects treated this month

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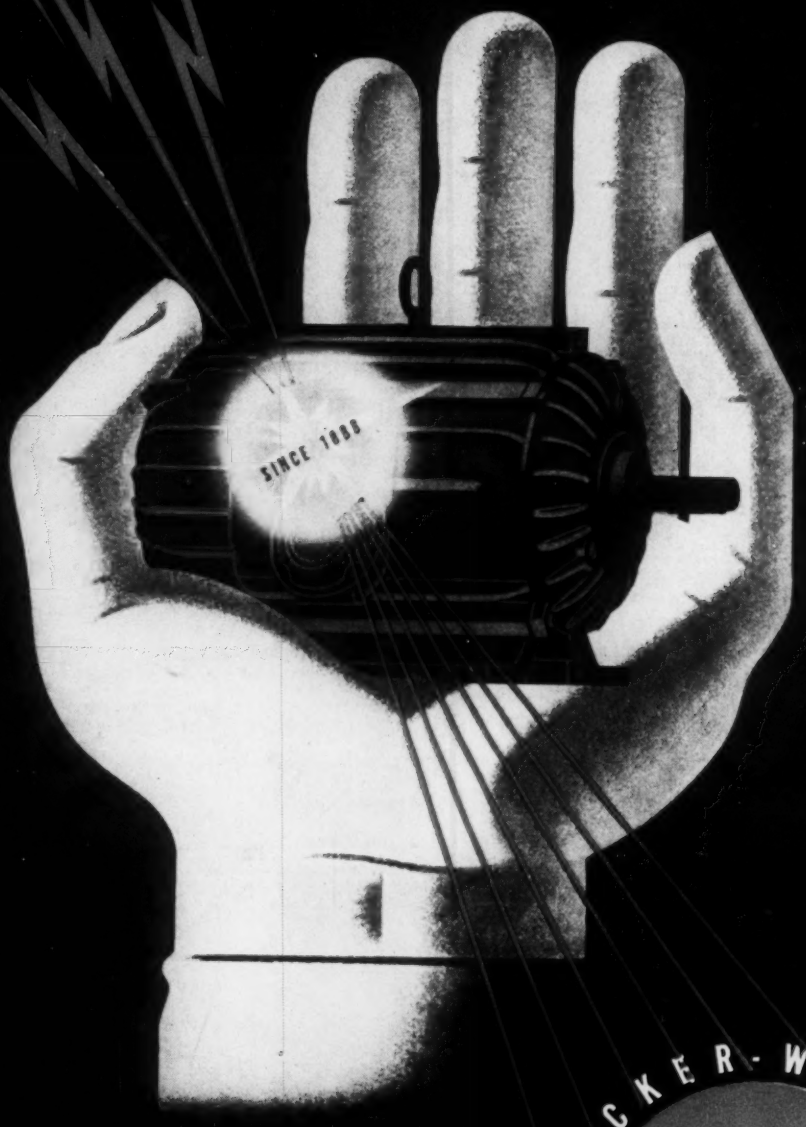
NEED MOTORS OR GENERATORS?



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The Editor's Pages

"ILL WIND" BLOWS AXIS NO GOOD

Air conditioning is an ill wind that is blowing the axis no good in the matter of American war production. Just how important air conditioning is for war output is reflected at the Package Machinery Co., which is completing re-tooling to produce essential items for the Navy.

Absolutely clean and cooled air is the first requisite for the assembly room, where tests will run continuously 24 hr a day. If the temperature of the room started at normal outside temperature, heat generated by the test work would shoot the thermometer up 25 deg in less than an hour without air conditioning.

PAW ISSUES FUEL OIL ORDER

Petroleum distribution order No. 13, replacing the provisions of WPB order L-56 dealing with limitations on delivery of fuel oil, was issued June 30 by Petroleum Administrator for War Harold L. Ickes. The new order incorporates the provisions of L-56 plus three important changes:

1) Fourteen new states and that part of Florida west of the Apalachicola river have been added to the territory in which deliveries of fuel oil are prohibited for consumption in new equipment installations or installations converted to fuel oil from other fuels. With this addition the order now covers the entire United States. Exceptions are provided for cases where certain conditions existed in the newly added territory.

2) Delivery of fuel oil for weed control purposes is prohibited.

3) Where new space heaters have been acquired under OPA ration order No. 9, the delivery of fuel oil is permitted.

MINERAL WOOL INSULATION "ESSENTIAL ACTIVITY"

Likelihood of unavoidable fuel shortages next winter has led the War Manpower Commission to declare the application of mineral wool for home insulation an "essential activity." Announcement of the decision, made in the interest of fuel conservation, is contained in a letter from the executive director's office to Wharton Clay, secretary of the National Mineral Wool Association. The letter says in part:

"The War Manpower Commission's committee on essential activities has interpreted the installation of mineral wool to be encompassed within building maintenance and repair services which is included with group 31, repair services, of the list and index of essential activities. Copies of the list and index have been forwarded to all United States Employment Service offices and to all War Manpower Commission area, state, and regional offices for use in connection with placement and stabilization policies and programs."

Workers in mineral wool manufacturing plants had previously been declared "essential" by the WMC, due in part to extensive requirements for insulation by the Army and Navy, says Mr. Clay.

AMENDED ORDER ON COIL OR TUBE ASSEMBLIES

Schedule III to the War Production Board's limitation order L-126, as amended June 26, presents required specifications for coil or tube assemblies for refrigeration condensers or coolers.

The required specifications established by the order do not prohibit, the order specifically states, "the use (in the production or fabrication of, or the delivery, acceptance, or installation of coil or tube assemblies for condensers, or coolers of the finned type, only where the tubing is expanded to obtain the bond between tubing and fins) of seamless steel tubing ⅝ in. OD size and larger, in coil or tube assemblies manufactured prior to September 15, 1943."

Welders at the plants of the York Corp. have put on the war paint literally in the battle of production to bring a new kind of totem art to the factory. Each of the grim "devil masks" is the individual work of its owner. One of the masks is shown here in use by a worker helping to weld the heavy steel plates of a General Sherman tank

OIL BURNER ORDER L-74 AMENDED

To restrict manufacture and secure more equitable allocation and freer distribution (especially to the armed forces) of existing stocks of oil burners, the War Production Board issued limitation order L-74 as amended on June 28, 1943. It is the purpose of the order to have all orders for class B burners routed through the plumbing and heating division of WPB so that prospective purchasers can be directed to available supplies.

Under the amended order, burners are divided into three classes. Class A comprises those burners which use No. 5, 6, or heavier fuel oil, or which are designed specifically for shipboard use or for heat processing. Class B comprises those burners which use Nos. 1, 2, 3, or 4 fuel oil, except a class C burner which is a pot type vaporizing burner using No. 1 fuel oil exclusively.

An approved order for a class A or C burner is one authorized under a construction order or for the account of a claimant agency. All orders for class A or C burners for other end uses and all class B burners must be approved on WPB form 2727.

The order prohibits placement of an order for any oil burner to be used in an area restricted by order L-56 or any order issued by the Petroleum Administrator for War, unless such order be accompanied by a copy of an authorization from PAW permitting the delivery of oil for such burner.

Two other WPB orders had a direct bearing on this amended order. No. L-79 restricted delivery of oil burners to ultimate consumers, except on orders bearing a preference rating of A-10 or higher. L-79 has now been amended to remove oil burners from its restrictions. Order L-56 prohibited the delivery of fuel oil to new installations of oil burners unless approval had been obtained from PAW.

LACKENS NAMED NIAA HEAD

Frederic I. Lackens, advertising manager of the Hays Corp., was elected president of the National Industrial Advertisers Association, Inc., at the central regional wartime conference of the NIAA held recently in Cleveland.

Vice presidents elected include: Alan Ballantyne, Novo Engine Co.; Walter A. Bowe, Carrier Corp.; Wilmer H. Cordes, American Steel &

The Editor's Pages

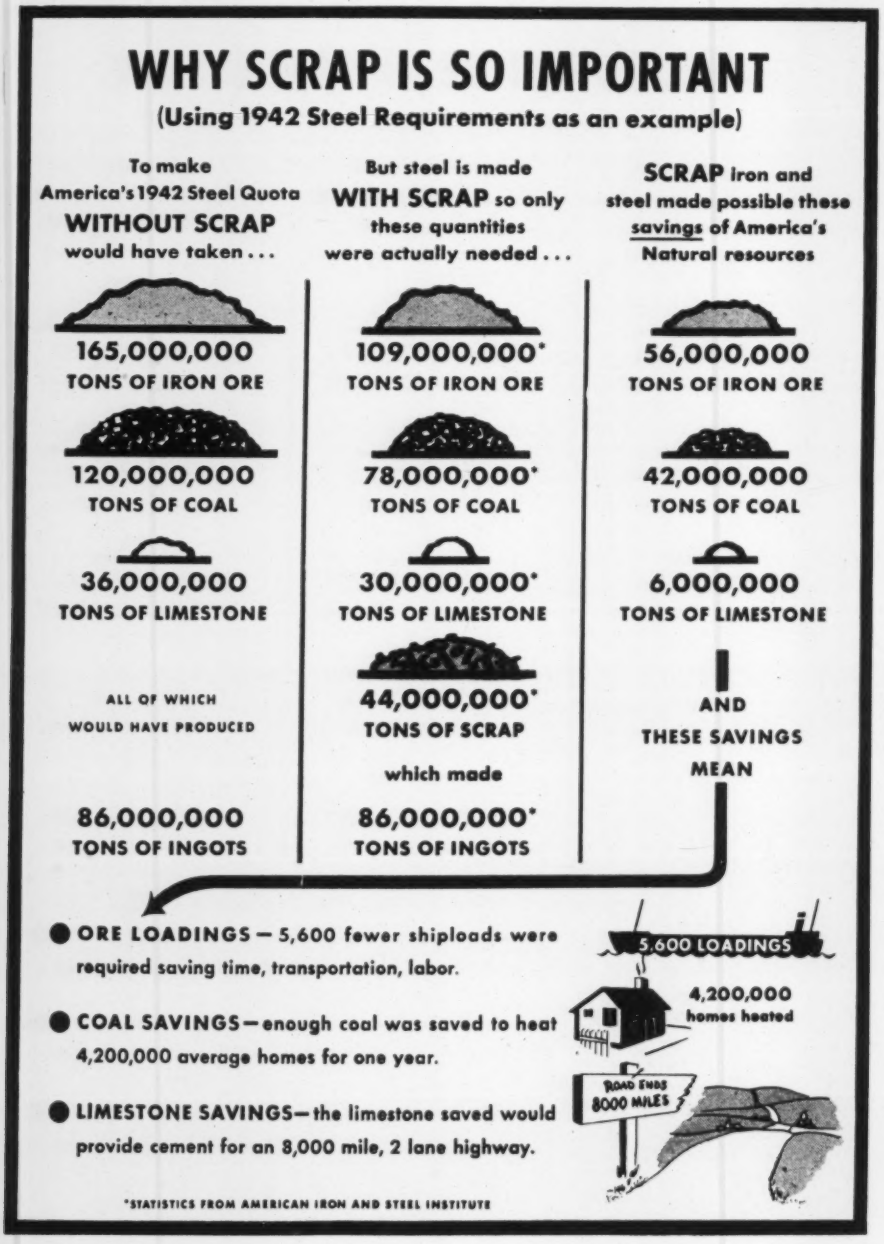
Wire Co.; Edward V. Creagh, American Chain & Cable Co., Inc.; John A. M. Galilee, Canadian Westinghouse Co., Ltd.; John H. Kunkel, Fluor Corp.; J. M. McKibben, Westinghouse Electric & Mfg. Co.; and Graham Rohrer, Baldwin-Hill Co.

This chart, issued by the Business Press Industrial Scrap Committee, graphically shows the sound logic back of the continual drive for MORE and MORE iron and steel scrap. Eloquently it tells why every bit of idle metal should be put in use, be sold to someone who can use it or disposed of as scrap.

The saving in raw materials, transportation and manpower made possible by the use of scrap is reason enough, in these war-tightened days, for everyone to pitch in and help the scrap program. But there are two bigger and more important reasons:

1. With scrap, steel can be made FASTER.

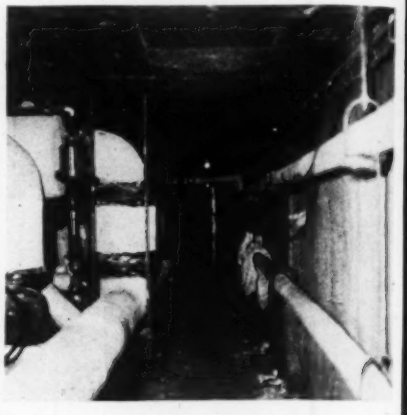
2. The use of scrap CONSERVES our nation's natural resources.


WHY SCRAP IS SO IMPORTANT
(Using 1942 Steel Requirements as an example)
To make America's 1942 Steel Quota WITHOUT SCRAP would have taken . . .
But steel is made WITH SCRAP so only these quantities were actually needed . . .
SCRAP Iron and steel made possible these savings of America's Natural resources
165,000,000 TONS OF IRON ORE
109,000,000* TONS OF IRON ORE
56,000,000 TONS OF IRON ORE
120,000,000 TONS OF COAL
78,000,000* TONS OF COAL
42,000,000 TONS OF COAL
36,000,000 TONS OF LIMESTONE
30,000,000* TONS OF LIMESTONE
6,000,000 TONS OF LIMESTONE
ALL OF WHICH WOULD HAVE PRODUCED
44,000,000* TONS OF SCRAP
which made
AND THESE SAVINGS MEAN
86,000,000 TONS OF INGOTS
86,000,000* TONS OF INGOTS
● ORE LOADINGS — 5,600 fewer shiploads were required saving time, transportation, labor.
5,600 LOADINGS
● COAL SAVINGS — enough coal was saved to heat 4,200,000 average homes for one year.
4,200,000 homes heated
● LIMESTONE SAVINGS — the limestone saved would provide cement for an 8,000 mile, 2 lane highway.
ROAD ENDS 8000 MILES
*STATISTICS FROM AMERICAN IRON AND STEEL INSTITUTE

DROP FORGE PLANT HAS PIPING TUNNEL

In a modern war plant built for 168 hr a week flow line production, the location of the piping presents a problem. The piping must not be in the way of cranes and other equipment, but must be readily accessible at all points for maintenance and alteration. Such was the case in the large drop forge plant of Kropp Forge Aviation Co. Joe E. White, project manager of the enterprise, describes how the problem was solved.

Requirements were met by installing the piping in a tunnel, with a



height of 6 ft 6 in. and the same width, lined with concrete. This tunnel has the form of a large capital T. The base of the T is at the power plant and the horizontal bar runs the length of the forge shop, a distance of several hundred feet.

The tunnel renders the steam, oil, and air piping accessible at all times for inspection, adjustments, or repairs.

AIR CONDITIONING CONTROLS POWDER BURNING

Air conditioning is playing a big part in insuring consistent performance of bombs and shells in actual combat, according to engineers of the Westinghouse Electric and Mfg. Co.

In rooms where powder is being loaded into the projectiles, it is extremely important that atmospheric conditions be carefully and consistently controlled. The burning rate of the powder varies considerably if the relative humidity of the loading rooms is subject to wide variation. It is also important, especially in winter months, to keep relative humidity at a constant level, sufficiently high to reduce the danger of explosions from static discharges.

* * *

HPAC's readers are cordially invited to comment on articles published in Heating, Piping & Air Conditioning. See pages 405-407 *this month for interesting additional information about subjects which have been treated previously.*

Heating Piping & Air Conditioning

with which is merged the Aerologist

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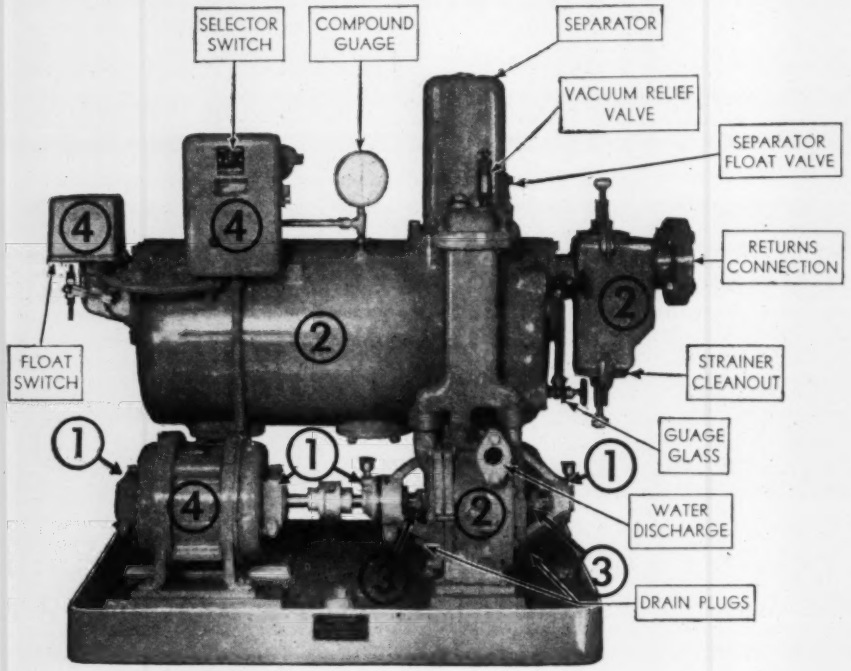
How to keep your Jennings Heating Pumps in First Class operating condition for Duration...



YOUR JENNINGS RETURN LINE VACUUM HEATING PUMPS

are an important part of plant or building equipment. They cannot be easily replaced and must last for a long time. Jennings Pumps are designed for long life, and have extra capacity which will count when it may be difficult to maintain lines and equipment free from leaks, but to give full service they must have proper care.

Now that these pumps have been in service all winter on the heating system, they should have a thorough inspection. Special attention should be given to the bearings, as no wearing contact occurs in a Jennings Pump as long as the ball bearings are properly maintained.

In most cases Jennings Pumps will require only the check-up we have recommended. If additional adjustments or repairs are found to be necessary, they should be undertaken at once. Do not wait until both time and materials are running short. Nash Service Branches are established throughout the United States. If you do not know the address of the one nearest to you, write directly to this office. Nash Service is yours to command, NOW.

SERVICE POINTERS

1 LUBRICATE PUMP AND MOTOR BEARINGS . . . Check bearings for end play, or radial play, and be sure they contain no grit, or foreign matter. If found to be clean and in good condition, fill with proper non-acid grease lubricant. If motor has oil lubricated bearings, flush out with fresh oil, and refill.

2 CLEAN STRAINER AND STRAINER SCREEN . . . Clean strainer well and strainer screen. If screen is found to be worn, and has passed foreign matter, thoroughly flush both receiving tank and pump casing. Drain plugs are conveniently located . . . Damaged screen should be repaired or replaced at once.

3 RENEW STUFFING BOX PACKING . . . Remove old packing and clean stuffing box. Fill with new loose-woven graphite-impregnated stuffing box packing, in individual rings. Be sure joints are staggered, and that ends butt properly without over-lapping. Do not finally tighten gland until pump is again in operation.

4 HAVE MOTOR AND CONTROL INSPECTED . . . Have an experienced electrical service man go carefully over motor and control. Follow his recommendations and if repairs or replacements are indicated, have the work done immediately.

5 CHECK OPERATOR'S INSTRUCTION BOOK . . . If operating instructions originally furnished with the pumps have been lost, write to us at once for a new copy for each pump. BE SURE TO FURNISH US THE TEST NUMBER OF EACH PUMP, as stamped on the pump name plate.

6 IF PUMPS REQUIRE REPAIRS . . . and you do not know the address of our nearest Service Branch, notify us at once, and we will have our Service Engineer get in touch with you without delay . . . There are well equipped Nash Service Branches in most principal cities, ready to give you every possible help.

NASH ENGINEERING COMPANY

254A WILSON ROAD • SOUTH NORWALK, CONN.

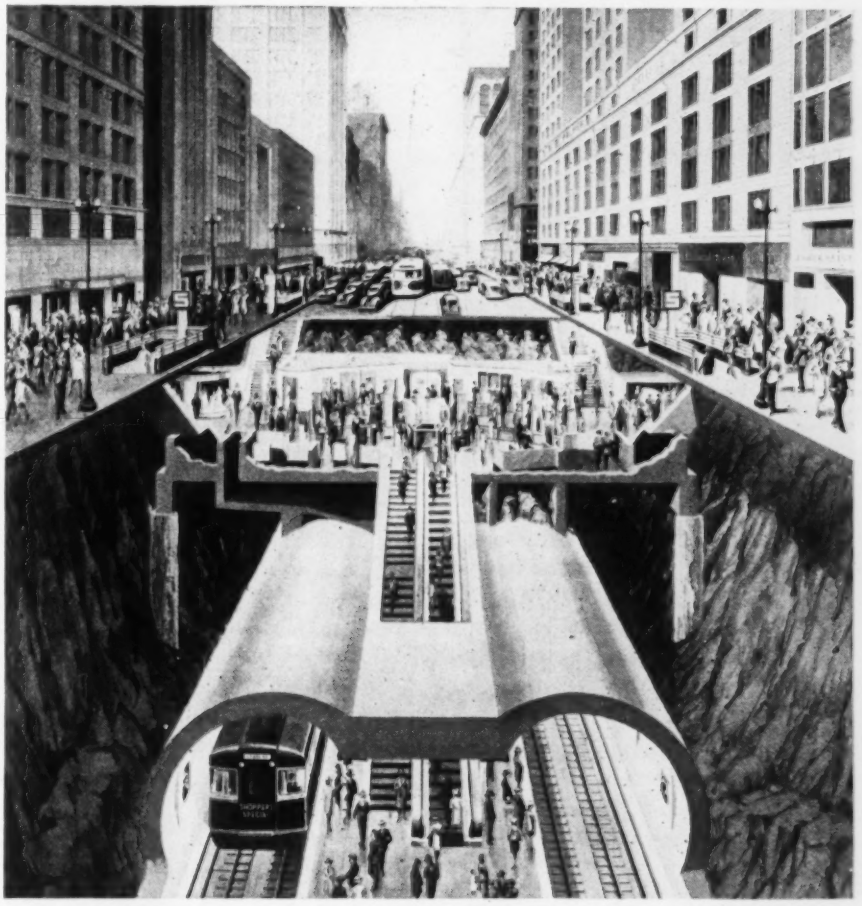
VENTILATING the new Chicago SUBWAY

Walter E. Rasmus, Subway Mechanical Engineer, Department of Subways and Superhighways, City of Chicago, Describes the Studies and Methods

THE CLYBOURN - Division - State street route of the initial system of subways for the city of Chicago is about ready to be placed in operation. At the present writing trains are making test runs within the subway. The basic, heavy construction of the Milwaukee-Lake-Dearborn street route of the initial system has been completed, but the equipping of this route will no doubt be delayed until after the war.

These subways have been designed and are being built by the department of subways and superhighways, city of Chicago, under the direction of Philip Harrington, commissioner, and Charles E. Deleuw, acting chief engineer, with the cooperation of the Public Works Administration, which is represented by the PWA subway commission, and Joshua D'Esposito, PWA project engineer, and their staffs.

In the design and building of this new subway system, the engineers have attempted to incorporate every modern feature for the safety and convenience of the traveling public. It was recognized that every effort should be made to provide underground travel which would be pleasant as well as speedy and convenient. It was also recognized that nothing would contribute more to the comfort of passengers within the subway than a first class ventilation system. In keeping with these objectives, available data on the ventilating facilities of the world's principal subway systems were studied, and a number of original investigations were conducted. Knowledge gained from these investigations is reflected in the standards of design that were adopted.



Artists' perspective of the subway structure at a downtown station

Heat Disposal

In the early days of subway and tunnel building, little attention was given to the ventilation of these structures. The accounts in the technical literature of the results of this omission and the struggles and efforts of engineers to correct the lack of ventilation in the early tunnels makes interesting reading. For instance, this sentence appears in a paper entitled The Great Alpine Tunnels, by Francis Fox, presented before the Royal Institution of Great Britain in 1901: "A heavy train of dining and sleeping carriages, with two engines conveying one of the crowned heads of Europe and suite, arrived at the exit of the Pracchia tunnel with both enginemen and both firemen insensible; and in other cases passengers have been seriously affected." Speaking of the St. Gothard tunnel, the author also states, "One might as conveniently travel in a furnace flue." Of course the problem in these early tunnels was greatly aggravated by the fact that coal burning locomotives poured out the products of combustion into the tunnels.

The advent of the electrically operated train made possible the elimination of smoke and combustion gases in tunnels, but did not eliminate the problem of heat. In the design and building of the earliest subways for electric train operation it was not generally recognized that the generation of heat by the car motors was a serious problem until these subways had been in operation for several years and the temperature of the subways was found to be on a continu-

Downtown station fan

ous rise. It was not uncommon for subway temperatures to get up in the 90's and stay there all summer, even though the outside temperature at various times dropped to the 60's or 70's. Since then there has been a great improvement in the ventilation of subways and it has been determined that the only practical and economical method of disposing of the heat produced by the car motors, other miscellaneous motors, the lighting system and by the passengers is by the circulation of outside air within the subway.

With this in mind, the designers of the Chicago subway made a careful determination of the amount of heat that would be produced in the subway and then provided sufficient ventilation so that the subway temperatures might be kept at comfortable levels. By far the greatest portion of heat generated in the subway will come from the car motors. The heat from lights, miscellaneous motors, and passengers probably will not account for more than 15 per cent of the total heat produced.

All of the electrical energy consumed by the car motors will be converted to heat principally by friction, and this heat will be soaked up by the interior subway surfaces and result in a continuous rise in temperature unless means for its removal are provided. The subway for the most part is at a depth of about 45 ft and the surrounding earth is an excellent heat insulator. It is estimated that under full operation, heat will be produced at the average rate of approximately 250 Btu per hr per lineal foot of tube. This is a very considerable quantity of heat as is at once evident when it is realized that it is equivalent to the burning of about 10 tons of coal in the Clybourn-Division-State street subway every 24 hr. The amount of ventilation required for heat disposal is actually much in excess of that required for supplying necessary fresh air to subway patrons. Thus the problem of the so-called ventilation of subways is not a problem of ventilation at all, in the ordinary sense, but is a problem of heat disposal.

Condensation

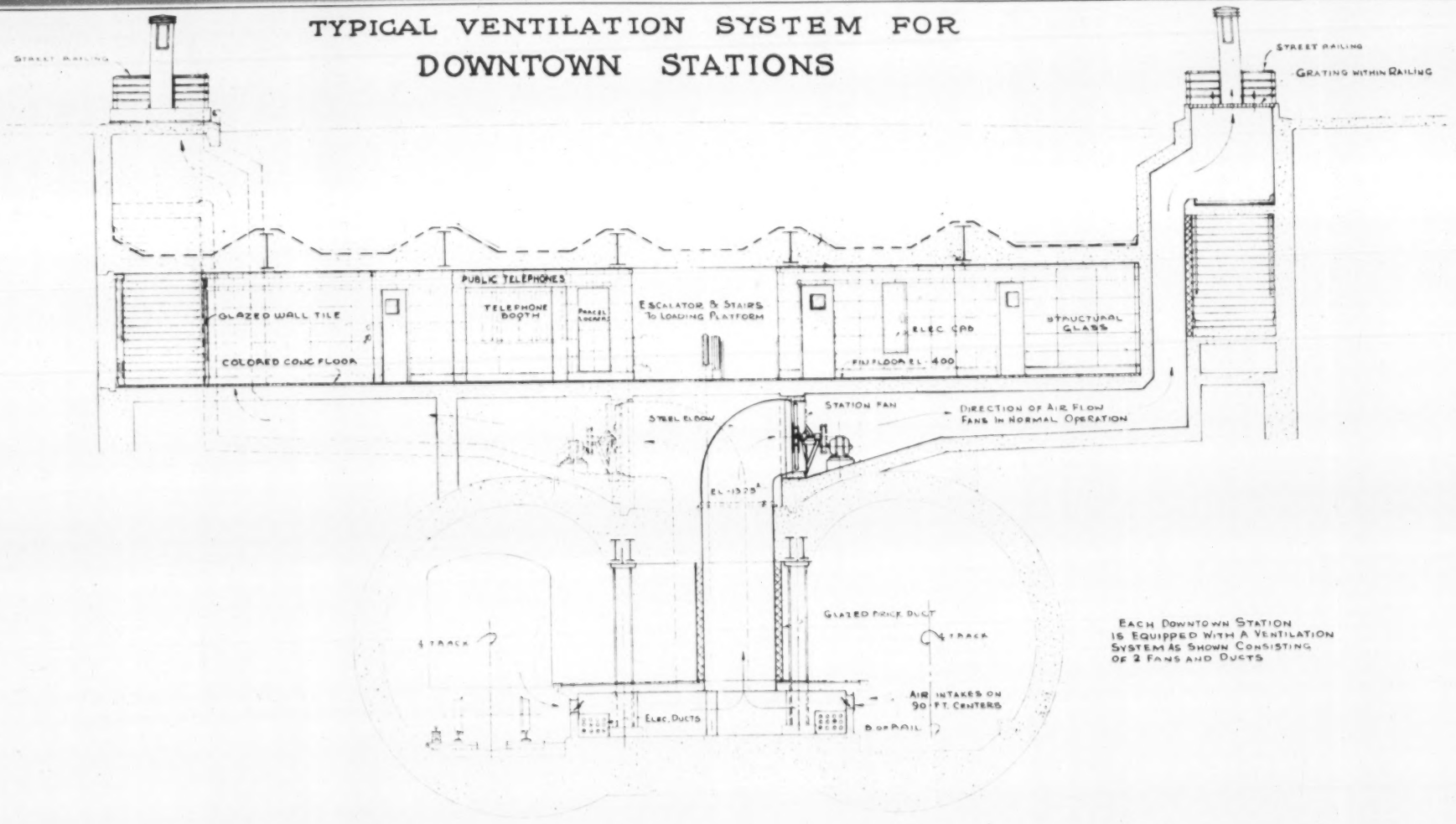
Another important function of a subway ventilation system is the elimination or reduction of condensation within the subway during the period preceding train operation. Since subways are not built in a day, this problem is present over a considerable period of time. As is well known, the temperature of the ground in Chicago at the depth of the subway is approximately 50 F throughout the year. Without mechanical ventilation, the subway tends to remain at this temperature throughout the summer with a rise of about 5 or 6 deg by midsummer as a result of the natural circulation of air by way of vent shafts, portals, and stairwells. Thus, during those periods when the dew point of the outside air is above, say, 55 F, all of the surfaces of the subway become bathed in condensation with serious corrosion of steel and electrical and mechanical equipment resulting. The critical period comes during the months of June, July, and August when the dew point of the outside air is seldom below 55 F. By operating all the fans on an average of about 10 hr per day during the month of June this year the subway temperatures have been raised to at least 62 F, and, in many locations, to levels approximating the outside temperatures. This raising of the subway temperatures has greatly reduced the number of periods of condensation and also the intensity of condensation during the few remaining periods when the dew point

Piston Action

Piston action of moving trains will be depended upon for normal ventilation of the major portion of the Chicago subway. Fans will be depended upon for ventilation of the tunnels during emergencies and for the ventilation of the tunnels at the river crossing. Fans will also be depended upon to supplement the piston action ventilation in the downtown stations between Lake street and Van Buren street. The electrically operated louvers will be depended upon to prevent short circuiting of air flows induced by fan operation and also for the retention of heat within the subway during freezing weather.

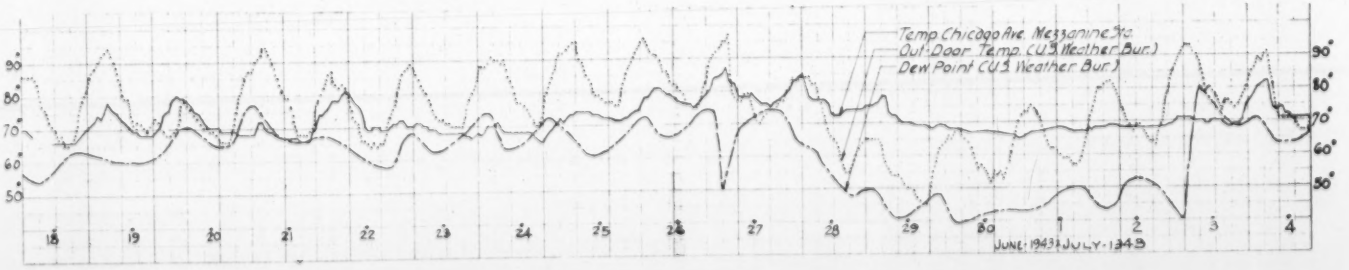
Emergency tunnel fan

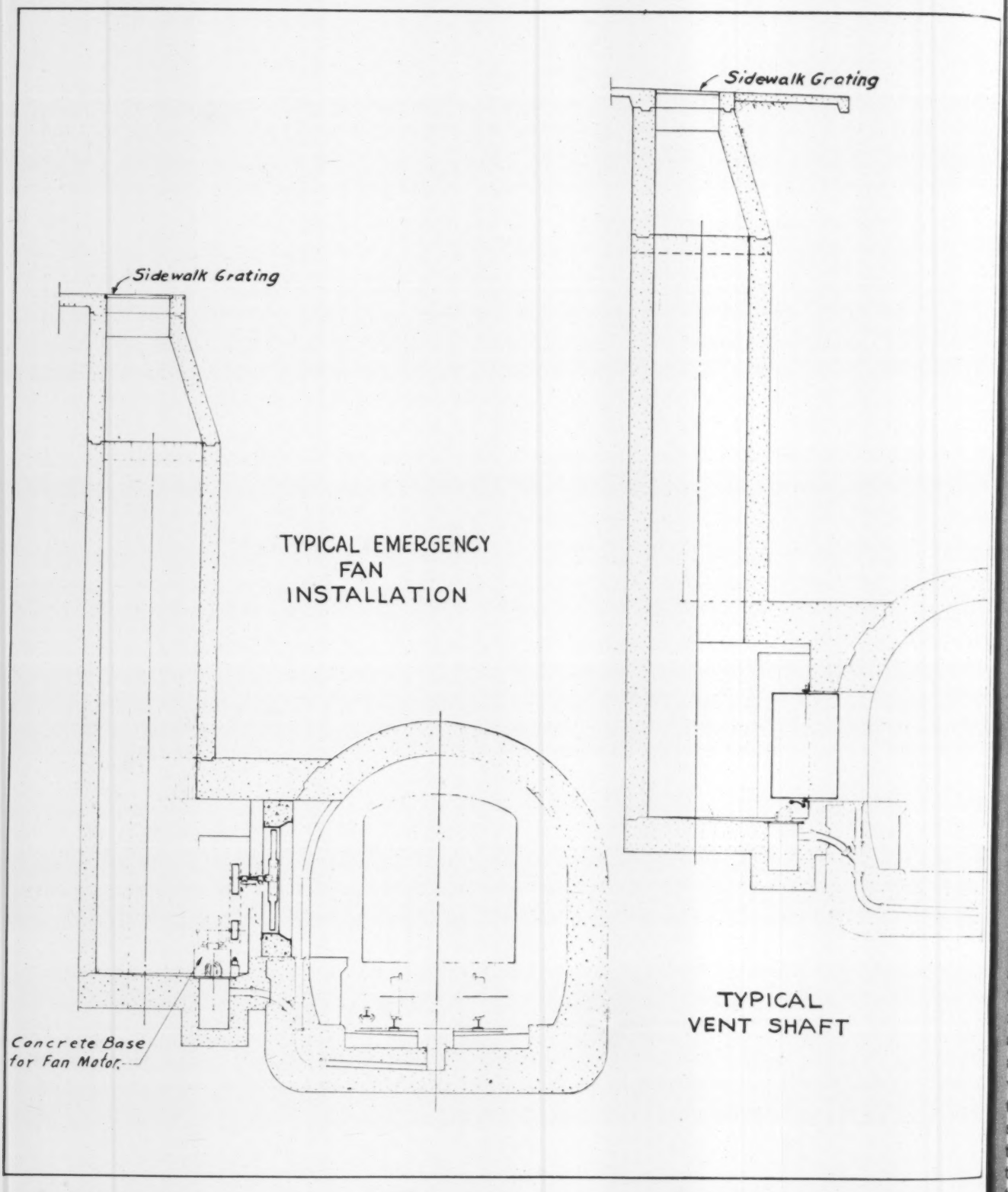
TYPICAL VENTILATION SYSTEM FOR DOWNTOWN STATIONS



EACH DOWNTOWN STATION IS EQUIPPED WITH A VENTILATION SYSTEM AS SHOWN CONSISTING OF 2 FANS AND DUCTS

TEMPERATURE CURVES FOR THE CHICAGO AVE. MEZZANINE STATION RECORDED BY RECORDING THERMOMETER





is above the raised subway temperature. It is confidently anticipated that continued fan operation will so reduce the frequency and the intensity of condensation that this problem need cause no further concern.

Starting on February 20, 1940, weekly observations of temperature and humidity in the subway were made at five different stations, these observations being continued to March 31, 1942. The averages of these readings were plotted on a graph together with outdoor temperature readings taken simultaneously with the subway temperature readings. Also plotted on this chart were the average temperature and dew point curves for Chicago. Of special significance in connection with the problem of condensation was the average dew points for Chicago. We will have condensation in the subway only at those times when the subway temperature is below the dew point temperature. The curves show, for example, that the subway temperature was constantly below the dew point temperature from May 6 until August 26, 1941.

When the temperature readings were commenced, the tunnels were being mined and constructed under an air pressure of from 10 to 13

psi gage. The high subway temperatures, between 72 F and 68F, found during the period from February 20 to May 21, 1940 were due to the heat resulting from the hydration of the setting concrete. About May 21, 1940, the connecting of the vent shafts to the tunnels was started and the subway temperature ranged between 66 and 61 F during the summer of 1940. By September of that year all of the vent shafts had been connected and the subway temperature began to vary with the variation in the outside temperatures.

In 1941, at no time was the subway temperature more than 8 deg below the dew point temperature and also for most of the critical period the temperature was less than 4 deg below the dew point. This indicated that it would take a comparatively small rise in subway temperatures to eliminate condensation troubles entirely. This has been verified by our experience with the operation of fans during the month of June of this year, when we succeeded in raising the temperature of the subway a minimum of 6 to 7 deg, practically solving the condensation problem.

About June 15, 1943, 12 recording thermometers were installed at strategic points in the Clybourn-Division-State street subway for the guidance of the operator of the fan and louver system. Plottings from the charts of the instrument at the Chicago avenue mezzanine station have shown that we have succeeded in raising the temperature of this mezzanine station up into the 70's. This is typical for all of the mezzanine stations which are running from 6 to 10 deg warmer than the tunnels which is to be expected inasmuch as the fan system is so arranged as to draw the warm outside air in through the stations and thence down into the tunnels.

Piston Action Ventilation

In ventilating a subway, the engineer has certain very powerful forces that will work in his favor if properly harnessed. The Chicago subways are single track tubes and the cross-sectional area of the trains is about ½ that of the tubes. Thus, moving trains will exert a positive piston action, pushing large volumes of air ahead of them and drawing large volumes of air in their wake. In partially closed tunnels these columns of air are several miles in length.

In order to get the maximum good from this more or less "free" ventilation, it is necessary to provide vent shafts from the tunnels to the surface at frequent intervals so that subway air may be forced out and fresh air drawn in. Now vent shafts are expensive structures as are all underground structures, so a careful study and investigation was made to determine the proper spacing to give the required results at a reasonable cost. This spacing of the vent shafts for the Chicago subways was found to be about 450 ft, using tunnel openings to the vent shafts whose area approximates 100 sq ft each. It is estimated that over two million cfm of air will be drawn into and discharged from the subway by piston action when traffic reaches its ultimate volume.

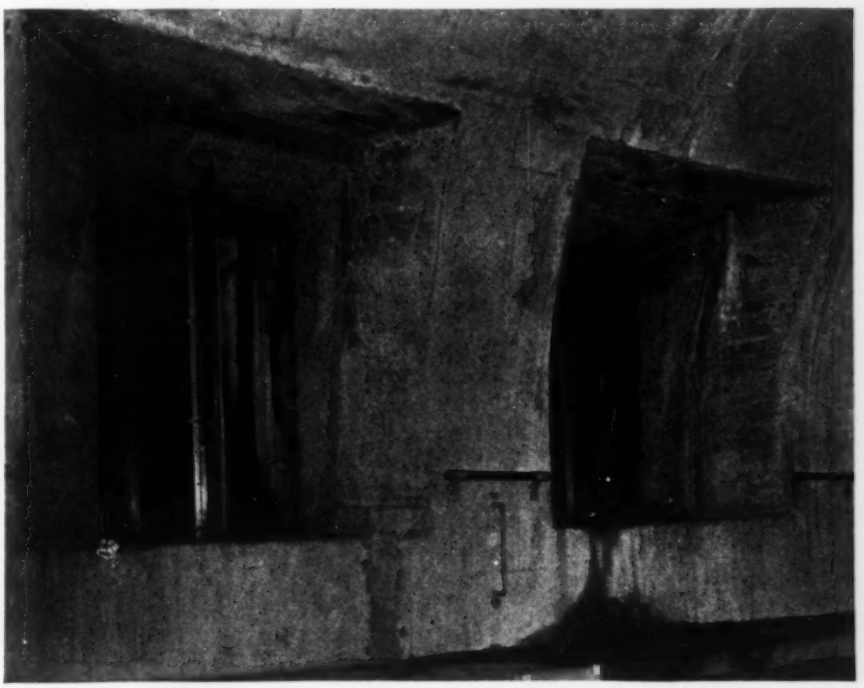
Vent Shafts

There are 69 vent shafts in the Clybourn-Division-State street subway, whose total length is about 5 miles. The vent shaft structure consists of a group of concrete tubes connecting the openings in the tunnel walls with a steel grating area at the sidewalk level. The circular tube construction was used because tubes can be built much more cheaply than rectangular structures at the depths encountered. Care was taken to provide sufficient grating area at the sidewalk level so that the passing of trains would not cause objectionable velocities of air under pedestrians who might be passing over them at the time of train passage. Care was also taken to so proportion the grating area that the entire width of a sidewalk would not be used, thus making it possible for pedestrians to avoid walking over the grating, if they so desire.

Blast Relieving Shafts

The passenger stations outside of the downtown area are equipped with vent shafts which have an additional purpose to that of providing ventilation—namely, the relieving of the blast action produced at the loading platforms by arriving and departing trains. The openings to these shafts are in the tunnels a short distance ahead of the point where the train enters the station and a short distance beyond the point where the train leaves the station. The openings ahead of the station total about 200 sq ft in area while those beyond the station total about 100 sq ft in area. The increased opening area at the entrance is required because the train enters the station at a higher speed than it leaves. Past experience has shown that without such blast relieving shafts the velocities of the air blasts on the loading

Vent shaft openings with louvers



platforms would be excessive and cause annoyance to patrons waiting to board trains.

These blast relieving vent shafts are not equipped with louvers inasmuch as they must always be open to perform their blast relieving function.

Emergency Ventilation

While piston action is extremely effective in moving air in and out of the subway when trains are moving, this action is, of course, absent when trains are stalled in the tubes as might happen in the case of fire or other accidents. It was therefore considered necessary to install mechanically operated fans to serve at such times, at least one such fan being situated midway between each pair of passenger stations. The vent shafts intervening between the fans and stations are equipped with automatic louvers or shutters which are so interlocked with the fan that when the fan is in operation the louvers are shut. Thus, the air is drawn in through the stations and sweeps the entire length of tube served by the fan.

Downtown Station Ventilation

Also, in the downtown area, it was necessary to supplement the "piston action" ventilation with mechanical ventilation by fans, inasmuch as the north and south bound tubes open onto a common loading platform giving a ratio of subway cross-sectional area to train cross-sectional area much greater than in the portions of the subway where the track is confined to a single tube. This fact, together with the fact that trains will make shorter runs at lower speeds in this area, reduces very materially the effectiveness of piston action.

The downtown station fans are 5 ft in diameter and are of the two speed reversible type with a capacity of 23,000 cfm each. In general, each downtown station, of which there are seven, has been equipped with two of these units. Under normal operation, these fans will draw air from the tubes through openings in the side walls of the station platform and discharge the air at street level. This results in fresh air being drawn in through the stair openings at the sidewalk level to the mezzanine station, through the mezzanine station, and down the stair and escalator wells to the tunnels. In reverse operation the fans will force fresh air through the openings in the side wall of the station platform into the tunnels, up through the stair and escalator wells to the mezzanine through the mezzanine station, and out to the surface through the entrance stairs. The fans will be operated in reverse only during cold winter weather when it will be desirable to circulate warm tunnel air through the mezzanine stations and stairways.

Typical downtown mezzanine station

Deep Tunnel Ventilation

At the river crossing, the subway is at its lowest depth; the depth is such that it would be uneconomical to sink vent shafts from the surface to take full advantage of piston action. One fan with a capacity of 65,000 cfm has been installed in each of the two tubes at this point to provide the necessary ventilation.

The diameters of the tunnel fans, including both the emergency and deep tunnel fans, range between 6 ft and 8 ft with capacities ranging between 50,000 and 85,000 cfm each. These fans are of the two speed type but are not reversible and will exhaust air from the subway, using the tunnel itself as an air duct.

These fans are mounted in the tunnel walls and require very little in the way of fan chambers. In fact these fans have been installed in vent shafts so that practically no structure cost is chargeable to the fans. This was an important consideration in the selection of the axial flow type of fan in preference to the housed centrifugal type, which would require separate fan chambers.

Remote Control

The fans and louvers will be remote controlled by the power supervisor through the supervisory control system. The subway is equipped with emergency alarm and telephone systems also terminating in the power supervisor's office. Thus, in case of fire or stalling of trains, word can be transmitted immediately to the power supervisor so that he may select and operate the proper fans for smoke removal or ventilation in the section of subway affected.

By observation of the charts from the recording thermometers, the power supervisor can determine which fans should be operated to keep temperatures in various sections of the subway at comfortable levels. It is anticipated that he will operate the fans at night for cooling during the hot portion of the summer and that during the cold weather in winter he will refrain from operating the tunnel fans, or at most operate them at slow speed, and keep a portion or all of the vent shaft louvers closed so as to prevent undue chilling of the subway.

The entire system is extremely flexible and is so arranged as to make it possible for one individual to control the temperatures and quality of air (within limits, of course) throughout the entire subway without leaving his office and

Table 1—Complete list of fans, State street subway

Fan No.	Diameter, Ft	Capacity, Cfm	Static Pressure, In. of Water	Installed Hp
EF-1	6'-0"	50,000	0.625	15
EF-2	6'-0"	55,000	0.625	15
EF-3	6'-0"	50,000	0.625	15
EF-4	6'-0"	50,000	0.625	15
EF-5	8'-0"	85,000	0.625	25
EF-6	8'-0"	85,000	0.625	25
EF-7	7'-0"	65,000	0.625	20
EF-8	7'-0"	65,000	0.625	20
EF-9	7'-0"	65,000	0.625	20
EF-10	7'-0"	65,000	0.625	20
EF-11	6'-0"	55,000	0.625	15
DTF-1	7'-0"	65,000	0.625	20
DTF-2	7'-0"	65,000	0.625	20
SF-1 to SF-13 incl.	5'-0"	23,000	0.625	7½

EF = emergency fans; DTF = deep tunnel fans; SF = station fans.

with the minimum expenditure of power.

Twenty-six Fans

The equipment for the mechanical ventilation of the Clybourn-Division-State street route consists of a total of 26 fans, of which 11 are for emergency ventilation of tunnels, two for ventilation of the tunnels at the river crossing, and 13 for the ventilation of the downtown stations. It also includes 44 sets of electrically operated louvers and 12 recording thermometers.

All of the fans are of the axial flow, non-overloading type, and are of very sturdy construction. Blades of aluminum or bronze were unobtainable because of the limitations on the use of critical materials during war time, so heavy gage steel blades specially treated for corrosion resistance were used. Each rotor is equipped with 14 blades and rotates within a heavy steel ring with an entrance cone. The fans are V belt driven by splash-proof electric motors, operating on 208 volt, 3 phase, 60 cycle current, and the shafts rotate in roller bearings. Special attention was given to the question of quietness of fan operation and to this end tip speeds were limited as follows:

Emergency fans ..... 14,000 fpm
Deep tunnel fans..... 13,000 fpm
Station fans ..... 9,700 fpm

The fan rings and entrance cones are embedded in concrete walls not less than 1 ft in thickness, giving a very solid support for the fan and eliminating the chance of noise resulting from vibration.

The station fans as installed are quiet in operation as would be expected with the comparatively low tip speed permitted for these particular units. The deep tunnel and emergency fans at high speed produce a considerable roar in the tunnels which will, however, be unnoticed when trains are moving in the subway. No objectionable noise even at high speed is apparent at the surface where these fans discharge.

Table 1 gives a complete list of the fans installed in the Clybourn-Division-State street subway together with sizes, capacities, static pressures, and installed horsepowers.

In most cases, four electrically operated louvers are required for each vent shaft and these are connected to each other by rods and cranks so that they may be operated by one operating mechanism. These mechanisms consist of a fractional horsepower motor direct connected to a gear reducer housed in a heavy cast iron case. The louvers themselves are equipped with flexible edges to prevent stalling by ice or debris in the openings. The louvers and frames are heavily constructed of steel, hot dipped galvanized after fabrication.

The ventilation equipment was installed by Geo. E. Mathis and Son, ventilation contractors. This company also fabricated the louvers and frames and other metal work in connection with the job.

LIME CLOGGING, SLIME ARE AIR CONDITIONING PROBLEMS

MY OWN experience with lime clogging problems has been chiefly with the relatively hard water in the limestone districts of Pennsylvania. In those areas the analysis indicates a general average of 7 grains total hardness per gallon, including calcium hardness of 70 to 80 parts per million, and magnesium hardness of 35 to 40 parts per million.

When waters of this type were passed through nozzles in our processing rooms at any temperature above 100 F a definite incrustation was made evident immediately. The accumulation would progress to the point of requiring physical removal of the deposit after each 30 days' operation. The difficulty was most severe when we were using a type of humidifier where the water was thrown by centrifugal force against a number of closely spaced atomizing teeth arranged around the periphery of a disc and over which the heated air was circulated. The maintenance program was to remove the clogged apparatus from the atmosphere and place it in an earthenware crock where the incrustation could be removed by a bath of dilute muriatic acid. Immersion for a period of 24 hr would loosen the scale so that it could be removed with a stream of water.

The obvious inconvenience of such a procedure caused us to try a number of expedients. No definitely desirable result was obtained from any of them until we installed zeolite water softening equipment. The controls on the softeners are so adjusted as to regenerate frequently enough to insure that the treated water will not exceed 2 grains hardness per gallon.

Slime accumulations in air conditioning systems are, of course, a constant problem. Those of our air washers which handle large amounts of air from tobacco processing rooms are subject to large deposits of this material and considerable difficulty is experienced with plugging of spray nozzles and piping. The obvious result is the delivery of much dry or otherwise untreated air.

To eliminate interruptions and unsatisfactory operation, we now use a chemical product developed for the purpose. This material is added daily and the washers now function for an entire week without stoppage. At the end of each week the entire system must be thoroughly cleaned.

Some caution is necessary in using any of these cutting compounds in that too large a concentration will tend to destroy the protective paint on the interior of the washer. Furthermore, it is wise to select some compound which is made up with a rust inhibitor to prevent accelerated corrosion which sometimes occurs with these water treatments. During the period when we were testing the products designed to prevent the rapid fouling due to slime, we made a number of tests with plates of unprotected and painted steel suspended in solution of several concentrations for accelerated tests.—RUSH D. TOUTON, technical director, Bayuk Cigars, Inc. Member of HPAC's board of consulting & contributing editors.

Importance of Air Conditioning to War Effort:

VITAL TO PRODUCTION

THE PROGRAM for removal and conversion to war work of existing refrigerating machines now installed for comfort air conditioning service has been in effect for some time. In the summer of 1942 a number of air conditioning equipment owners, headed by Lessing J. Rosenwald of the conservation division of the War Production Board, arranged a meeting and offered their equipment to the war effort; in November, 1942, the WPB used equipment branch made it quite specific that wherever practicable, consideration be given to the use of existing equipment.

This program was prompted at the time by the vital necessity for refrigeration in the manufacture of synthetic rubber; a large tonnage of refrigeration was needed in a short period of time to meet the requirements of this program and the facilities for manufacture could not possibly have built the equipment in the time allowed. The sizes of machines for the requirements of synthetic rubber manufacture as well as the processing of the product have been greater than 100 tons in most every case. At that time the WPB arranged with the owners and operators of large air conditioning plants for the release of various components of their plants for use in the war program and before any specifications were prepared or purchases made for new refrigeration, the WPB used equipment branch had to be consulted.

The Office of Price Administration, in maximum price regulation No. 136 as amended, provided that the maximum price applicable to the sale at the direction of the War Production Board of any industrial or commercial air conditioning or refrigerating equipment to the manufacturer of such equipment by a person using such equipment and not engaged in the business of selling such equipment, shall be the installed cost of such equipment to such user, less depreciation at the rate of 5 per cent per year; and that the maximum price applicable to the resale of any such equipment by the manufacturer on a reconditioned and guaranteed basis to a new user designated by the War Production Board shall be the total cost of the equipment to the manufacturer, including the cost of inspection, dismantling, and shipment to the manufacturer's plant, estimated if necessary; plus the cost of reconditioning the equipment, of shipment to the purchaser, and of installation, estimated if necessary; plus the amount of any other out-of-pocket expenses incurred in connection with such sale, estimated if necessary; plus a reasonable charge for engineering and a reasonable profit, not upon the cost to the manufacturer but in the light of the manufacturer's risk and responsibility; *provided,* that the amount of such profit together with a description of the transaction, has been submitted to the Office of Price Administration, Washington, D. C., for its review and that the Office of Price Administration has approved or failed to disapprove the amount of such profit within 30 days after receipt of the information. (Am. 55)

Dramatic Evidence

A number of department stores and other buildings have released part of their comfort air conditioning equipment for use in war production — particularly in connection with the synthetic rubber program. The removal of a large refrigeration machine from a building where it is wanted—and is needed in normal times—for vital wartime production is a very practical—almost dramatic—evidence of the importance of air conditioning and refrigeration in industry and commerce.

Those owners who have released part of their equipment for war use are to be commended for their patriotic action, as are the manufacturers who have cooperated with the government's program.

The owner's net cost of the equipment was broken down into a schedule of items:

Cost to owner of machine less drive and speed control.

Cost to owner of drive and speed control.

Cost to owner of freight, cartage, and rigging.

Cost to owner of labor of installation.

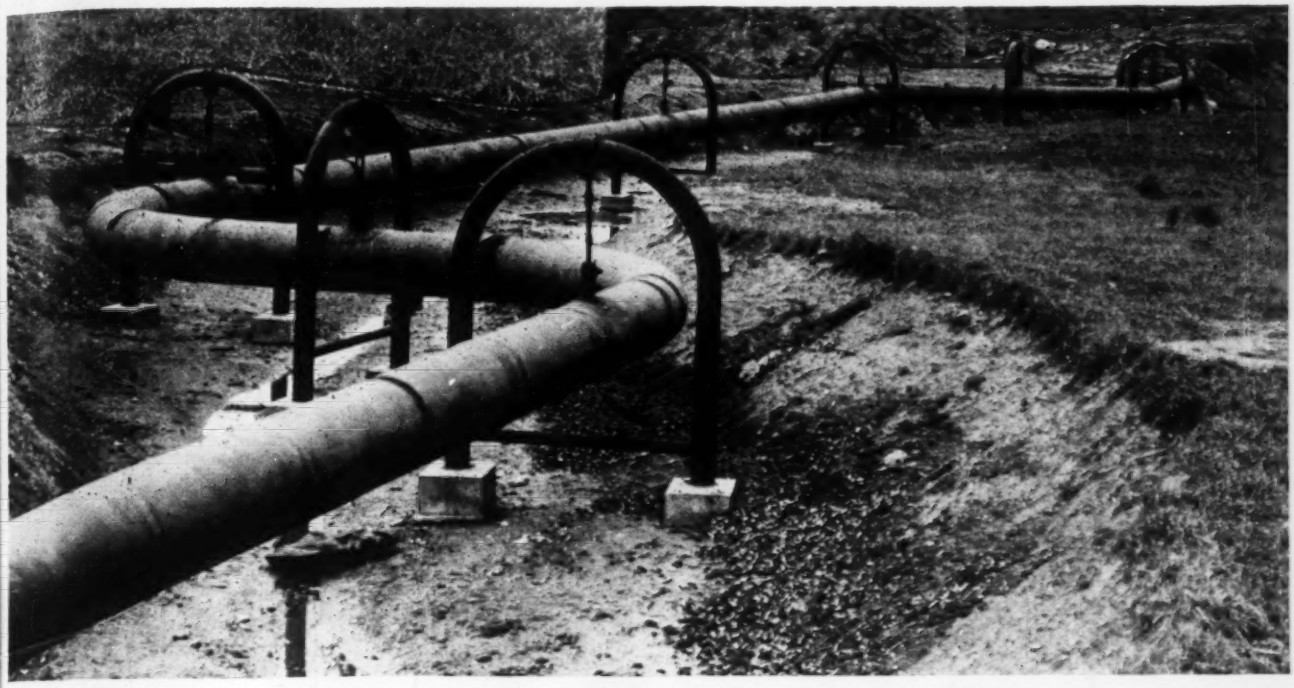
Cost to owner of such accessories as pressure gages, safety controls, water regulating valves, etc.

Cost to owner of such wiring, piping, etc., that would be destroyed or could not be repaired or restored.

Cost to owner of cutting, patching, decorating, etc. that was directly chargeable as a part of the original cost to the owner of installing the items being moved.

The total of the items less 5 per cent per annum for depreciation represents the sum to be paid the owner for the above.

The synthetic rubber program required the immediate installation of refrigerating systems for chilling water used in the process of making the crude synthetic from the basic elements. Since temperatures of 40 to 60 F are required, this fits practically with the performance of the machines used for comfort cooling and quite a number of machines were removed and reinstalled for this purpose. The crude basic synthetics, like natural rubber, require rolling or compounding and it was well known that the heat generated in the operation is sufficient to burn up the synthetic unless an ample quantity of refrigerated water is supplied. For processing natural rubber it is seldom necessary to use water lower in temperature than can be supplied by cooling towers, rivers, or other natural water sources. Therefore, refrigerating machines have been allocated by WPB for processing the crude synthetic rubber. Other conversions of comfort cooling equipment have been made for industrial processes in general, such as airplane engine assembly plants, special calibrating and testing.



Typical steam transmission line under construction

The Design of STEAM TRANSMISSION PIPING

Arthur McCutchan Explains Factors to Consider in Choosing Between Large or Small Pipe Size

WHILE IT is not possible to establish hard and fast rules for the design of steam transmission lines—because each line must be tailored to fit the particular service conditions—there are a number of problems common to all such lines. These problems involve determination of pipe size and wall thickness, selection of insulation, calculation of line losses, provision for thermal expansion, and the like. The author has had occasion to consider these problems in the design of a number of steam transmission lines for proposed steam sales projects and for several long heating and process steam lines.

The purpose of this article is to discuss the factors entering into the solution of these inter-related problems and to present tables which facilitate the design of steam transmission lines. In order to demonstrate the use of these tables two lines—a normal pressure loss, and a high pressure loss steam transmission line—are assumed and the conditions responsible for the use of each are described.

Normal Pressure Loss Line

It is assumed that saturated steam at a pressure of 100 psi gage is required for process heating. The maximum estimated steam demand is 200,000 lb per hr with an average load of 100,000 lb per hr. The permissible pressure at the delivery end of the line under zero flow conditions is given as 120 psi gage because of pressure limitations on the process equipment. Superheated steam at 300 psi gage and 700 F is assumed to be available in a central station power plant situated about one-quarter of a mile from the point of usage. It is proposed to supply saturated steam to this line at 120 psi gage pressure by means of a pressure-temperature reducer to be installed in the central station.

In order to simplify presentation, questions of returning condensate and possible generation of power are not considered in this article. Of course, the use of return condensate, if available in satisfactory condition, would be carefully considered, as would the economic possibilities of substituting a back pressure turbine for the reducing valve, but the source of the 120 psi saturated steam is only incidental to the design of the transmission line.

As far as the line is concerned, the important factors from the conditions described in the foregoing are that the pressure is to be controlled from the inlet end of the line, that under maximum steam flow of 200,000 lb per hr the pressure drop in the line should not exceed 20 psi, and that under the no-flow condition the pressure at the discharge end should not exceed 120 psi gage.

High Pressure Loss Line

Again it is assumed that saturated steam at 100 psi gage is required for process heating. The maximum estimated steam demand is 200,000 lb per hr as before, with an average steam requirement of 100,000 lb per hr.

In order to reduce heat loss in the transmission line and expansion difficulties, a desuperheater is assumed to be installed at the central station power plant. Allowing a pressure drop of 10 psi through this desuperheater, the condition of the saturated

steam leaving the central station would be 290 psi gage, 420 F. The difference between this pressure and the 100 psi gage required at the discharge end of the line is available to force steam through the transmission line. Since the average steam flow is 100,000 lb per hr, while the line must be designed for a maximum flow of 200,000 lb per hr, a pressure reducing valve at the discharge end of the line must be provided.

The reduction in pressure taking place in the transmission line and in the reducing valve tends to superheat the steam, but the energy required to produce the higher velocity at the downstream end of the line and the heat loss in the line both offset this tendency. The final heat content of the steam will be checked in connection with the calculation of heat loss of the line, but for preliminary purposes it can be considered as at saturation temperature.

Determination of Pipe Size

The permissible pressure drop for a maximum steam flow of 200,000 lb per hr, from the conditions given for the normal pressure loss line, is 20 psi. The straight line distance is 1400 ft, hence the drop per 100 ft for a line using slip joints to provide for expansion would be 20/14, or 1.42 psi.

Using the average pressure in the line of 110 psi gage and a steam flow of 200,000 lb per hr the nearest available pipe size is found from Fritzsche's chart [see Piping Handbook, by Walker and Crocker] as 16 in. For schedule 30, 0.375 in. wall, 16 in. pipe—which is suitable for the pressure-temperature condition — the drop per 100 ft is found as 0.7 psi or about one-half the permissible drop. The drop for a 14 in. pipe would be slightly greater than the assumed drop and would leave no margin for extra line length and equivalent length of bends, loops, etc., used to secure flexibility. However, if slip joints are to be used consideration should be given to changing the initial or final steam conditions sufficiently to use the 14 in. size rather than the 16 in. which would be satisfactory for all the various types of line construction considered herein.

The straight line distance between the central station power plant and the point of usage was assumed to be 1400 ft. The permissible drop per 100 ft for the high pressure loss line using this straight line distance can be determined directly. Allowing a 10 psi margin for drop through the reducing valve at the outlet end, the available pressure drop to force steam through the transmission line is 290 —110, or 180 psi. The drop per 100 ft would be 180/14, or 12.85 psi.

Referring to a Fritzsche pressure drop chart, a preliminary estimate of the pipe size may be obtained based on the average pressure in the line of (290 + 110)/2, or 200 psi and the maximum steam flow of 200,000 lb per hr.

The nearest available pipe size is found as 8 in. For schedule 40 pipe, which would be suitable for the pressure-temperature condition, the pressure drop per 100 ft would be 10 psi. Since the total pressure drop for the 1400 ft is 140 psi, it is apparent that an 8 in. pipe installed above the ground using directional changes, expansion bends, or offsets also might prove adequate if the total equivalent length does not exceed about 1800 ft. The amount of extra pipe required to provide for thermal elongation of the line is considered in a later section of this article.

Large or Small?

The solution of an engineering problem usually lies in the choice of the proper method from several possibilities. In the design of steam transmission piping, a fundamental question is that of a small diameter line with comparatively high pressure drop as against a larger diameter line with a smaller drop. That's the subject so ably covered here by Mr. McCutchan, engineer, engineering div., The Detroit Edison Co.

Problems involved in the design of steam transmission lines are discussed in this article with reference to a so-called "normal pressure loss" and a "high pressure loss" line. Each line is designed to accomplish the same purpose; that is, to supply the same quantity of steam at approximately the same final condition.

Which type of line is finally selected for a given installation will depend upon a number of factors —particularly cost—which are not considered in this article, but the apparent advantages of the smaller high pressure loss line should be given careful consideration. The advantages as developed here are less tonnage of pipe, elimination of condensate loss during normal operation, reduction in heat loss of line, and reduction in forces on anchors. The number of expansion loops or bends required to absorb thermal expansion of the smaller, slightly higher temperature line is the only apparent disadvantage over the larger normal pressure loss line. Even this disadvantage is offset by the reduction in cost and in the overall heights of bends required.

The discussion of the method of providing for thermal expansion of these two types of transmission lines given in the second installment of this article to be published next month, will be carried through in detail in order to illustrate the use of the expansion bend tables. These tables (to be presented next month) are believed to fill a need for a means of selecting conventional expansion bends with a minimum of computation.

Check for Critical Velocity Limitation of Flow

Since the pressure desired at the discharge end of the high pressure loss line is considerably less than 50 per cent of the initial pressure, the possibility of the acoustic velocity* limiting the maximum capacity of the transmission line should be taken into account. The critical velocity may be computed from the expression:

$$v_c = 60\sqrt{gnPV}$$

in which v_c = critical velocity at outlet end, fpm; n = adiabatic constant = 1.13 for saturated steam; P = absolute pressure at outlet end, lb per sq ft; and V = specific volume at outlet end, cu ft per lb.

While it is not difficult to determine the critical velocity from this formula, the corresponding critical pressure can be found from Table 1 by simply forming the ratio

$$\frac{w \text{ (lb per hr of steam)}}{a \text{ (cross sectional area of pipe, sq in.)}}$$

If the critical pressure found from Table 1 is below the discharge pressure of the line, there is no danger of a limiting velocity being obtained.

In the case of this particular transmission line, $w/a = 200{,}000/50 = 4000$. The corresponding critical pressure of 29 psi gage is well below the 100 psi gage discharge pressure of the line, thus indicating that the critical velocity will not be attained.

Determination of Pipe Wall Thickness

In connection with the determination of pipe sizes for the normal pressure loss line and the high pressure loss line, statements were made that schedule 30, 0.375 in. wall, pipe was suitable for the 16 in., 120 psi line while schedule 40, 0.322 in. wall, pipe was selected for the 8 in., 290 psi line.

The Code for Pressure Piping requires that outdoor steam distribution piping on the premises of an industrial establishment shall conform to the requirements of section 4 of the code on district heating piping systems, provided the pressure does not exceed 600 psi or the temperature exceed 750 F. For higher pressures and temperatures such piping is required to conform to section 1 on power piping systems. Steam distribution piping within the buildings of industrial establishments also is required to conform to the power piping section.

Wall thicknesses are determined from the formula.*

$$t_m = \frac{PD}{2S} + C$$

in which t_m = minimum wall thickness, in. = 87½ per cent of nominal wall thickness; P = maximum internal pressure, psi; D = outside diameter of pipe, in.; S = allowable stress

*How to Design Steam Piping for Maximum Capacity and High Pressure Drop, by Max W. Benjamin, Heating, Piping & Air Conditioning, September, 1936, pp. 475-477.

Table 1—Pressure corresponding to critical velocity (Saturated Steam)

$\frac{w}{a}$	Psi Vacuum	Psi Gage
0	15	
1,000	4	
2,000		7
3,000		18
4,000		29
5,000		40

Values for Table 1 have been taken from design chart No. 12, Tube-Turn catalog and data book No. 111, p. 219.

Table 2—Allowable *S* values of pipe in power and district heating systems

	ASTM A53, A106		ASTM A135*	
	Grade A Seamless	Grade B Seamless	Grade A Resistance welded	Grade B Resistance welded
Minimum ultimate tensile strength	48,000	60,000	48,000	60,000
150 F	12,000	15,000	10,200	12,750
250 F	11,520	14,400	9,800	12,250
350 F	11,040	13,800	9,400	11,700
400 F	10,800	13,400	9,150	11,500
450 F	10,560	13,200	8,950	11,200

*Where electric resistance welded pipe is subjected to supplementary tests to demonstrate that the weld is equal to the minimum tensile strength specified for the pipe material, the *S* values given for seamless pipe may be used.

at operating temperatures, psi (see Table 2 of this article for values applicable to usual steam transmission piping); and C = allowance for corrosion, etc., = 0.065 in. for power and district heating piping.

Heat Loss of Line and Final Condition of Steam

The amount of line heat loss and quantity of condensation, if any, will vary with the steam velocities, thickness of insulation, weather conditions, and whether the lines are installed in a conduit or tunnel, or above ground.

Assuming that the lines are installed above ground and insulated with 2 in. of 85 per cent magnesia insulation, or equivalent, and wrapped with tar paper or otherwise weatherproofed, the heat loss may be estimated as follows:

	16 In. Line	*8 In. Line*
Steam temp., F	350	420
Average air temp., F	45	45
Temp. difference	305	375
Btu per hr per sq ft of pipe per F*	0.25	0.25
Btu per hr per sq ft of pipe	76.3	93.7
Sq ft of surface per linear ft	4.20	2.26
Btu per hr per linear ft	321	212
Btu per hr for 1400 ft line	450,000	297,000
Including allowance of 25% for bends, etc.	560,000	372,000
For average flow of 100,000 lb per hr, Btu per lb.	5.60	3.72
For maximum flow of 200,000 lb per hr, Btu per lb	2.80	1.86

*Heat Transfer through Insulation in the Moderate and High Temperature Fields, by L. B. McMillan, Transactions, American Society of Mechanical Engineers, Vol. 48, 1926, p. 1277.

The heat drop required to accelerate steam from an initial velocity corresponding to 290 psi gage, 420 F to a final velocity corresponding to 100 psi gage and approximately 338 F may be found from the following basic relation:

$$\text{Btu per lb} \times 778 = \frac{v_2^2 - v_1^2}{2g}$$

in which 778 = mechanical equivalent of heat, ft lb per Btu; v_2 = velocity at outlet end of line, fps; v_1 = velocity at inlet end of line, fps; g = acceleration due to gravity = 32.2 fps per sec.

For the 16 in. normal pressure loss line, the heat drop required to accelerate steam from initial to final velocity under a flow of 200,000 lb per hr amounts to only 0.15 Btu per lb and hence is negligible. The difference in velocities for the high pressure loss line is significant, as shown by the following calculation of heat drop required to accelerate steam in the 8 in. line:

$$v_2 = \frac{200{,}000 \times 3.88 \times 144}{3{,}600 \times 50} = 620 \text{ fps}$$

$$v_1 = \frac{200{,}000 \times 1.50 \times 144}{3{,}600 \times 50} = 238 \text{ fps}$$

$$\text{Btu per lb} = \frac{(620)^2 - (238)^2}{64.4 \times 778} = 6.5$$

Heat content of the 290 psi gage, 420 F steam at beginning of line = 1203.0 Btu per lb. Heat loss in 8 in. line for 200,000 lb flow = 1.86 Btu per lb. *Heat drop to accelerate steam = 6.50 Btu per lb. Total drop in line = 8.36 Btu per lb. Therefore, heat content of steam leaving line = 1194.64 Btu per lb.

Since the heat content of saturated steam at 100 psi gage is 1189.7 Btu per lb, it is apparent that under maximum flow conditions there would be no line condensation and the steam would have about 8 deg of superheat. For the average steam flow of 100,000 lb per hr the superheat would be increased because the heat drop to accelerate steam is reduced and this more than offsets the increased heat loss in the line per pound of steam flowing. For practical purposes the steam can be considered as dry and saturated at the point of usage. When warming up the line and for light loads provision must be made for removing condensation, but under normal operation there will be no drips to remove from such a line.

In the case of the 16 in. normal pressure loss line the condition of the steam leaving the line is found as follows:

Heat content of the 120 psi gage, 350 F steam at inlet end of line = 1192.4. Heat loss in 16 in. line for 200,000 lb per hr flow = 2.80 Btu per lb. Heat drop to accelerate steam 200,000 lb per hr = 0.15 Btu per lb. Therefore, computed heat content of steam leaving line = 1189.45.

Since the heat content of saturated steam at 100 psi gage is 1189.7 Btu per lb, it is evident that the result just computed will not be obtained but that the steam leaving the line will correspond to the dry saturated condition. The difference between 1189.45 and 1189.7 or 0.25 Btu per lb will be provided by heat of condensation of the drips which form. For this maximum flow condition of 200,000 lb per hr and a heat of condensation of 880 Btu per lb, the pounds of condensation is found as follows:

$$\text{Line condensation} = \frac{200{,}000 \times 0.25}{880} = 57 \text{ lb per hr}$$

For the average flow condition of 100,000 lb per hr, the line condensation is determined by dividing the product of the steam flow and the difference between the computed heat content of the steam at the discharge end of the line and the heat content of saturated steam at the leaving pressure by the heat of condensation.

The heat content of the 120 psi gage, 350 F steam entering the 16 in. line was 1192.4 Btu per lb. The heat loss in this line was computed as 5.60 Btu per lb for an average flow of 100,000 lb per hr. The heat drop to accelerate steam is insignificant

[*Concluded on p.* 404]

*Note: If this line discharges into a lower velocity line, there will be a corresponding reconversion into heat energy. If the velocity at the point of usage is equal to the initial velocity of steam entering the transmission line, the only loss of heat will be that due to radiation and convection in the line.

SAM LEWIS' PAGE

By Samuel R. Lewis
Consulting Mechanical Engineer
Member of HPAC's Board
of Consulting & Contributing Editors

• On this page, Mr. Lewis comments informally on a few of the problems he has met and is meeting in his wide and varied experience as a consulting engineer doing heating, piping, and air conditioning work. Some of the time you will agree with him . . . some of the time you won't . . . but you'll always find him interesting and thought-provoking.

He Who Buries a Return Pipe

The other day I found a vacuum heating pump trying to become a water supply system. The centrifugal vacuum pump in a big church seemed to be weary and very reluctantly would return the condensate. The fireman used to be troubled with more water coming into the boilers than was leaving them in the form of steam. After months of unsuccessfully trying to find a leaking city water supply valve, a broken vacuum return buried under the concrete basement floor was discovered. This pipe had been run through a water bearing gravel and sand deposit. The pump, innocently and dutifully, had been doing its best to drain the water and sand out of the deposit until the sand had worn down the close clearances between the impeller and its casing.

He who buries a return pipe or permits any client to bury one in such manner as to defy easy inspection should be disbarred.

Steam Units Save the Day

All of us are more or less afflicted by the necessity for compliance with various local, state, and federal codes. I have helped to write many of these and have some appreciation of the good which they accomplish. I have also felt oppressed and indignant at various times when compelled to make some serious change in design because of conflict with a forgotten or misunderstood code.

Recently, I designed a very large heating-cooling plant in Wisconsin. Refrigerated water, to the equivalent of melting around 6000 tons of ice in every 24 hours, is pumped to the air cooling convectors. The same convectors and piping are used with the same water as a heat transporter in winter, except that in winter the water is of course heated.

There is a state code provision that hot water shall not be used as a heat transport to convectors which heat outside air, because of alleged danger that the water may freeze in cold weather and thus, by preventing ventilation, cause a health menace. I was compelled therefore to provide enough steam heated convectors in the plant to heat a minimum amount of outside air, though I was permitted to use outside air in the 60 remaining hot water heating apparatuses.

This plant now has been through both winter and summer extreme conditions. We found when there was a sudden sharp, cold snap that in so large a system we could not heat the great volume of water in mains and convectors quickly enough to prevent freezing, but that the well distributed steam heated units, which could keep the interior of the building above freezing temperature and which could be heated quickly even at remote distances, saved the day.

How often in this life do we build better than we know!

Conversion to Coal

I have been having a heavy dose of surveying existing boiler plants with a view to their being changed so as to burn coal rather than oil. It is interesting to note the degree of skill that has been developed by various engineers, architects, and owners in locating boilers so that they are unreachable by anything but liquid fuel.

I have been surprised, too, at the number of plants — not particularly juvenile in appearance—which never have burned any fuel but oil. Comparisons of cost between coal burning with automatic stokers and bunker C oil generally do not indicate any saving by using oil. In most cases the change was made from coal to oil because of need for greater output.

In public buildings (particularly schools), however, the labor item is significant in cost considerations. In one recent survey of a district having six school buildings, it appears that coal will cost $8.50 per ton and that extra help to get the coal in and the ashes out will cost $6.25 per ton, not counting the cost of gasoline and tires and automobile maintenance for the peripatetic assistant's transportation between buildings.

Two-Way Stretch

Here is a testimonial to the Hartford Steam Boiler Inspection and Insurance Co.

I have been working harder and faster than my capacity due to trying to keep ahead of the war production demand. I designed a long, crooked, temporary, high pressure steam main with a low pressure return main alongside it for heating an emergency research laboratory.

As in most war work, the budget was close and the authorities could not settle down in their minds as to what sort of a job of underground work they would compel me to design. After making the drawings and estimates over three or four times, successively less attractively, I became discouraged and tired and careless and when finally they decided to run both mains in a split tile I paid little attention. The contract then was let and the mains were welded into place. The supply and return mains were placed side by side. There are several right angle turns. Hartford (who insures them) then told me what I had neglected; that expansion of the supply will cause it to move more than the return main alongside it. The difference in length when hot will cause such dissension between the two at the right angle turns despite their apparently safe anchors and expansion loops, as to render them an unacceptable insurance risk.

Of course I was happy—and very much relieved—to cooperate by seeing to it that the possibility of dissension between these parallel pipes was eliminated. Now if the steam main stretches five inches and if the return main stretches only three inches, there remains ample clearance between them at the movable right angle turns.

Steam Transmission Piping

[*Concluded from p.* 403]

for this low flow, so can be neglected. The computed heat content of the steam leaving the line is therefore 1192.4—5.6=1186.8 Btu per lb. The difference between this computed heat content and that of saturated steam at 100 psi gage of 1189.7 Btu or 2.9 Btu per lb must be supplied by heat of condensation.

For the average steam flow of 100,000 lb per hr, the amount of condensate is found as follows:

$$\text{Line condensation} = \frac{100{,}000 \times 2.9}{880} = 330 \text{ lb per hr}$$

Provision must be made, of course, for removing the much greater quantity of drips which forms during warming up the line and for small steam flows.

"OPEN FOR DISCUSSION"

(Comment from HPAC's readers on HPAC's articles)

Standardized Heating and Ventilating Equipment for Fighting Ships

THIS SECTION has read with considerable interest the article in the July HPAC by Commander Urdahl and Lieutenant Everetts covering the standardization program sponsored and so successfully carried on by the air conditioning section of the Navy bureau of ships covering heating, ventilating, and cooling equipment for shipboard use under their jurisdiction. This section heartily endorses such an article and wishes to point out that the standardization program definitely was and is the outstanding factor enabling the fan and blower industry to meet the required rate of production of such equipment for our fighting ships. If such a program had not been instituted, the industry would have been unable to meet the demands for the shipbuilding program and consequently the air conditioning section, Navy bureau of ships, under Commander Urdahl deserves a tremendous amount of credit for putting into operation and carrying out so successfully such a program.

The simplification of the production problem, while only one of the several distinct and outstanding benefits of the standardization program, was of paramount importance to this section, which is responsible for the overall production and operation of the industry.—W. H. HOLT, chief, fan & blower section, general industrial equipment division, War Production Board.

Decentralized Air Conditioning Serves Big Aircraft Factory

I HAVE just read Mr. Hendrickson's article on the Douglas plant at Long Beach, published in the July HPAC.

Naturally, to break away from tradition as we did for this plant, we had many an interesting argument for and against the unitary design.

The original decision on the blackout problem was most interesting. At that time, October of 1940, the use of windows and skylights with the provision of blackout coverings held ready for use was first suggested. For a factory building with the traditional sawtooth roof with monitor sash, it would require a considerable tonnage of galvanized steel sheets held in reserve at strategic points ready for prompt application.

The cost of this item was estimated and then the structural engineers made some comparative figures on the two types of roof construction, saw tooth versus flat, and it was found that the elimination of the monitor sash with their steel frames, glass and operating mechanisms and the simpler steel roof truss construction would make a sizable saving in the cost of the buildings themselves and this saving could be applied to the air conditioning systems.

After the decision was made for the completely windowless plant, the next question was central plant versus unitary installations. Mr. Hendrickson has set forth many of the points discussed. I might add a few more.

As my office also provided the plans for the underground piping, sanitary sewers, storm drains, domestic water mains, fire mains, gas mains, compressed air mains, and also the building plumbing and fire sprinkler systems, we concluded right in the beginning that to add another system of water mains which would distribute cold water for cooling and hot water for heating from a central plant to a well spread out group of buildings on a very large site would add to the problem of obtaining proper grades for the sewers and storm drains. With wide and completely paved streets between the buildings, it was a difficult enough problem to get required grades for proper drainage with the various crossings of the above mentioned service pipe lines.

We had available an ample supply of utility produced electric power at reasonable rates, so I could see no advantage of adding a high pressure steam plant to generate electricity or to operate turbine driven centrifugal compressors. In fact, I never advocate putting a client into the power business unless there are very definite advantages over purchased power.

I am quite sure that the figure given by Mr. Hendrickson that the central plant installation with circulated water would average 15 to

20 per cent more in first cost than the Long Beach unitary installation, is well on the conservative side. It is common practice to provide all underground concrete pipe tunnels, both between buildings and in buildings, in the concrete work of the general contract work. If the cost of such pipe tunnels is added to the air conditioning contract, the difference in cost will be increased.—O. W. OTT, consulting mechanical engineer.

I HAVE read the article in your July issue by Mr. H. M. Hendrickson, Decentralized Air Conditioning Serves Big Aircraft Factory, with considerable interest. There are several points on which I would like to make comments.

On the subject of decentralization vs. centralization, I cannot see how there can be much argument regarding the air handling equipment, whatever the method of refrigeration may be. This is pretty well limited by building conditions, particularly space for equipment and ducts, and practice in these plants has been rather similar in size of equipment used—whether direct expansion or central chilled water. For that reason I do not quite understand Mr. Hendrickson's distinction, except as it applies to the source of the refrigeration.

I am a little bit surprised at his statement that in southern California the systems must change from heating to cooling very rapidly—often heating in the morning and cooling in the afternoon. Considering a winter design condition of 40 F outside temperature for the locality, and the fact that this is a windowless building with a substantial internal load, it seems to me that cooling is required the great majority of the time. This may not necessarily mean operation of the compressor, but it is difficult to see why heating is needed very often. In any event, if reheat is required simultaneously with cooling, separate piping must be furnished to carry it whether the system uses direct expansion or chilled water as the source of cooling effect. To say that reheating is a virtue available only to a unitary system gives a misleading impression; it can obviously be used with any kind of system, and is more often than not used as an adjunct to central systems to promote good zoning.

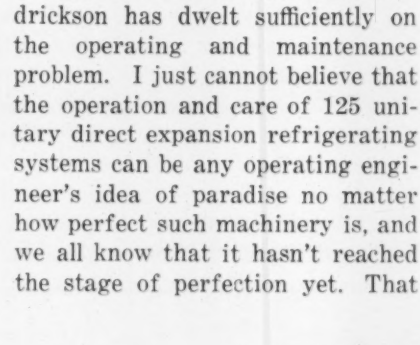
It is probable that the centralized chilled water refrigeration system is somewhat higher in first cost than a multiplicity of direct expansion equipment, but, as Mr. Hendrickson points out in his article, the economies possible with a centralized plant, particularly when steam is available for turbine drive, will in most cases result in a standoff in the operating costs (excluding maintenance and supervision) for the two types of systems. Incidentally, his figure of 40 F chilled water temperature seems to be on the low side. As he has pointed out, internal latent load is very light, and water temperatures of 45 F and even as high as 47 or 48 F are practical.

I do not believe that Mr. Hendrickson has dwelt sufficiently on the operating and maintenance problem. I just cannot believe that the operation and care of 125 unitary direct expansion refrigerating systems can be any operating engineer's idea of paradise no matter how perfect such machinery is, and we all know that it hasn't reached the stage of perfection yet. That is a tremendous number of pieces of reciprocating machinery spread all over the plant. It involves 125 motors to inspect and oil, 125 starters to service, 125 sets of V belt drives, a like number of pressure switches, and several times 125 expansion valves to keep in operation. Mere routine attention and inspection in itself must be an enormous problem, not to mention servicing and attention to leaks in the refrigerant piping system.

Obviously the entire problem of operation and maintenance is immensely simplified by the use of a few large centrifugal refrigeration machines, using low pressure refrigerant, as has been done in most of these plants which have been air conditioned.

It is to be greatly hoped that comparable operating records are being kept for the centralized and decentralized plants, and that some day they will be available for publication—since it is only by such actual results that the factors of power consumption, steam consumption, refrigerant loss, routine service and maintenance labor costs can be determined.—W. A. GRANT, director of application engineering, Carrier Corp.

How Much Oil Did Rationing Save?



MR. HECKEL'S analysis of the fuel oil rationing program in the July HPAC is very interesting and checks rather closely with the experience in other offices. Robert Gray, who served in a capacity similar to Mr. Heckel's on a local board in Long Island, also made an analysis of the rationing experience there which agrees substantially with Mr. Heckel's results.

The overall experience of the rationing program resulted, I believe, in the achievement of a saving in the order of slightly more than 33⅓ per cent of the normal fuel oil consumption, whereas Mr. Heckel's analysis shows a saving of about 27 per cent. The difference may be due to the increased value of the period three, four, and five coupons in the middlewest which was not the case in the more critical areas.

When it is realized that last year, the rationing program got off to a late start and that the public was rather confused by divergent opinions published in the papers and did not take full advantage of conservation measures, the results of the program are extremely gratifying. This year, with stress being laid on the importance of

insulation and easy means available through the Federal Housing Administration to finance conservation measures, the possibilities of further savings are apparent.

Efficient utilization and more widespread use of insulation and storm sash should effect savings this year of something like 40 per cent with less overall discomfort and no real hardship.—ROBERT K. THULMAN, mechanical engineer, technical division, Federal Housing Administration.

What's the Baby's Name?

TAKING YOU at your word (see p. 145, back section, July HPAC), the following comments outline my opinion as to the proper terminology to be employed in discussing the subject of "radiant heating":

Mr. Mills has made a good point in underlining the somewhat confused conditions existing in the terminology applied to the art of "panel heating," etc. Many years ago the Byers Co., when it first became interested in this work, cast about on its own hook for the proper descriptive term and finally decided that "radiant heating" was the most accurate. We reasoned that since this system of heating was based on an entirely different physical principle from conventional systems, it could best be described by using the term "radiant" which in turn is a form of the word "radiation." Frankly, we know of no other term which so aptly fits this system of heating since it is exactly a process of radiating heat from a warm body to another one. Or if you prefer to look at it the other way around, it is a system whereby heat loss from the body to other surfaces is prevented by bringing up the temperature of those other surfaces to a place where the heat loss through radiation from the body is at a comfortable level. From this it became obvious to us that the word "radiation" was the key word and that any description of this heating system was meaningless unless some derivative of the word was used.

We realized, of course, that for a time there would be some possibility of confusion between "radiator" and "radiant." But we felt that what was originally a misnomer—since most radiators are really convectors—could not be corrected, and that little would be gained by resorting again to improper nomenclature simply to avoid a remote possibility of confusion. In other words, two "wrongs" would hardly make a "right"—if one "wrong" had already been made, why add to the confusion by making another one?

Over the past years we also considered such terms as "panel heating," "panel warming," etc., but discarded any idea of using these because we felt that they were not truly descriptive. Another objection is that they tend to imply a number of things—such as prefabrication, ready assembled heating units, etc. —which are not necessarily the case.

I sincerely hope that the foregoing comments will help you throw some light on this confused picture and I do want you to know how we feel—that until a more accurate term comes along, the baby's name is still "radiant heating."—C. A. HAWK, JR., engineering service dept., A. M. Byers Co.

Big Inch Pipe Line Completed

Workmen at Phoenixville, Pa., finish the final joint in the Big Inch pipe line which carries oil from Texas to the East. The welding was watched by an assemblage of government officials and oil and steel men

Acme

How to Size

RADIATORS AND CONVECTORS

When Used in the Same Hot Water Heating System

By F. E. Giesecke

IT HAS been found by experience that it is difficult to secure a balanced hot water heating system if convectors and radiators are used together in the same system. The reason for this difficulty is that the heat emission of a radiator varies approximately as the 1.3 power of the water to air temperature difference, and the heat emission of a convector varies approximately as the 1.5 power of that temperature difference. Consequently, if the temperature of the water circulating in the heating system is modulated with the varying heat requirement so that the radiators will, at all times, emit the required quantity of heat, and if the sizes of the radiators and convectors are selected so that each will emit the required quantity of heat when the heat requirement is high, the convectors will emit an insufficient quantity of heat when the heat requirement is low.

In the accompanying chart the two sloping lines show (approximately) the heat emission of radiators and convectors, in Btu per hour per square foot of equivalent direct radiation, for various water to air temperature differences. According to this chart, if the temperature difference (water to air) is 145 F, radiators and convectors will each emit heat at the rate of 240 Btuh per sq ft EDR; however, if the temperature difference is 50 F, radiators will emit heat at the rate of 60.1 Btuh per sq ft EDR, and convectors at the rate of 48.6 Btuh per sq ft EDR.

Examples Show Effect

The following examples illustrate the effect of the difference in the variation of heat emission by radiators and convectors with varying water to air temperature differences.

Example 1: Assume a building having two rooms exactly alike, each having a heat loss of 7200 Btuh when the indoor temperature is 70 F and the outdoor temperature is zero, one of the two rooms to be heated by means of a radiator and the other by means of a convector, and the radiator and the convector to be in balance when the outdoor temperature is zero, i.e., when the heat demand of each room is 7200 Btuh. Assume that for this condition the mean water temperature in the radiator and in the convector is 215 F, so that the temperature difference, water to air, is 145 F.

For this temperature difference, the heat emission of the radiator and of the convector will be, according to the chart, 240 Btuh per sq ft EDR, hence both radiator and convector should be of the sizes rated as 7200/240 or 30 sq ft EDR; in that case, both radiator and convector, when filled with 215 F water, will emit heat at the rate of 30 x 240 or 7200 Btuh, and the heating system will be balanced.

When the outdoor temperature rises from zero to 60 F, the heat requirement of each room decreases to one-seventh of 7200, or from 7200 to 1029 Btuh. The heat emission of the radiator must then be reduced to 1029/30 or 34.3 Btuh per sq ft EDR; to effect this reduction in heat emission, the water to air temperature difference must be reduced, according to the chart, from 145 to 32.4 F, and the mean water temperature in the radiator must be reduced from 215 to 70 + 32.4 or 102.4 F. With water at this mean temperature of 102.4 F flowing through the convector and the corresponding temperature difference, water to air, of 32.4 F, the heat emission of the convector will be, according to the chart, at the rate of 25.4 Btuh per sq ft EDR, or at the rate of 762 Btuh for the 30 ft convector, instead of at the required rate of 1029 Btuh. Consequently, the heat emission of the convector will be about 26 per cent too low when the outdoor temperature is 60, although it is in balance with the radiator when the outdoor temperature is zero.

Example 2: Assume the same conditions as in example 1 except that the size of the convector is selected so that it will be in balance with the radiator at the mean outdoor temperature instead of at the minimum outdoor temperature; that is, in this case at an outdoor temperature of 35 F.

With an outdoor temperature of 35 F, the heat requirement of each room will be one-half of 7200, or 3600 Btuh, and the radiator must emit heat at the rate of 3600/30 or 120 Btuh per sq ft EDR. For this rate of heat emission the temperature difference, water to air, must be (according to the chart) 85.1 F; that is, the mean water temperature in the radiator must be 70 + 85.1 or 155.1 F. With this water to air temperature difference the heat emission of the convector will be, according to the chart, 107.9 Btuh per sq ft EDR and, consequently, the required size of the convector will be 3600/107.9 or 33.4 sq ft EDR.

With a convector of this size, the heat emission will be 33.4 × 240 or 8016 Btuh when the outdoor temperature is zero, and 33.4 × 25.4 or 848 Btuh when the outdoor temperature is 60 F. Consequently, the heat emission of the convector will be (8016—7200)/7200, or over 10 per cent too high when the outdoor temperature is zero, and it will be (1029—848)/1029, or almost 18 per cent too low when the outdoor temperature is 60 F.

It appears from the foregoing cal-

Difficult to Balance

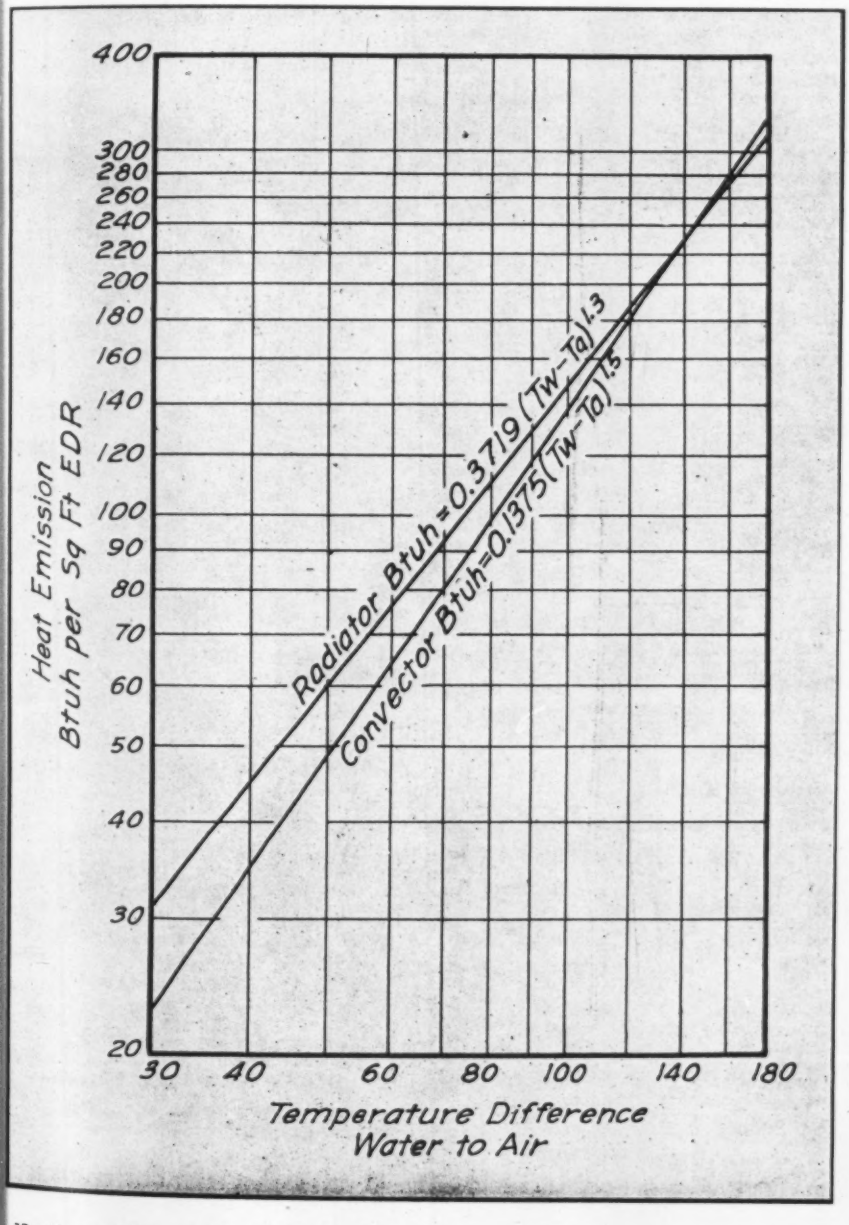
It's difficult to balance a hot water heating system when both radiators and convectors are used in the same circuits, for the heat emission of a radiator varies with the temperature difference between water and air according to one power and the heat emission of a convector varies in accordance with a different power, as indicated on the chart shown here. Dr. Giesecke —professor emeritus, Agricultural & Mechanical College of Texas, and a member of HPAC's board of consulting & contributing editors —explains the problem and gives two examples of it.

He suggests sizing the radiators and convectors so they will be in balance when the outdoor temperature is slightly below the mean outdoor temperature for the heating season. Thus, the heat emission of the convectors will be somewhat high during colder weather and somewhat low during warmer weather.

culations that when convectors and radiators are to be used together in a hot water heating system that they should be sized so that they will be in balance when the outdoor temperature is slightly below the mean outdoor temperature during the heating season; if this is done, the heat emission of the convectors will be from 12 to 15 per cent too high during the coldest period of the heating season and from 12 to 15 per cent too low during the warmest period of the heating season.

These calculations based on the chart are not accurate for all radiators and all convectors because the heat emission of radiators and convectors varies somewhat with the general forms and proportions of the radiators and convectors; in the case of radiators, the rate of heat emission may vary with a power of the water to air temperature difference differing slightly from 1.3; in the case of convectors, the rate of heat emission may vary with a power of the water to air temperature difference differing materially from 1.5. This method of calculating sizes of radiators and convectors for joint use can be safely applied if it is known definitely how the heat emission of the particular radiators and convectors to be used varies with the water to air temperature difference.

The heat emissions of radiators and convectors vary according to different powers of the water to air temperature difference. T_w is the temperature of the water within the radiator or convector, and T_a is the room air temperature, in degrees Fahrenheit



The formulas Btuh $= 0.3719\ (T_w - T_a)^{1.3}$ for radiators and Btuh $= 0.1375\ (T_w - T_a)^{1.5}$ are based on the definition that the heat emission of radiators and of convectors is at the rate of 240 Btu per hr per sq ft of equivalent direct radiation, when the temperature difference, steam to air or water to air, is 215 F — 70 F or 145 F, and on the assumption that the heat emission of radiators varies as the 1.3 power of that temperature difference and that the heat emission of convectors varies as the 1.5 power of that temperature difference. The exponents 1.3 and 1.5 are given in the 1943 ASHVE Heating, Ventilating, Air Conditioning Guide, p. 265; they are based largely on experimental determinations at the University of Illinois. However, it has been found that the exponent 1.3 may vary slightly with the type of the radiator and that the exponent 1.5 may vary materially with the type of the convector; consequently, the exponents 1.3 and 1.5 are recommended only as good average values to be used in the absence of exponents determined experimentally for the particular radiator or convector under consideration.

URGES CONSUMERS TO ORDER OIL NOW

The Petroleum Administration for War said last month that although the program for supplying heating oil this winter contemplates the filling of consumer tanks as soon after July 1 as possible, it may be that the tanks of many consumers cannot be filled until late summer.

Delay in filling some tanks, Deputy Administrator Ralph K. Davies said, will be due both to the necessity of utilizing fuel oil tank trucks in the most efficient wartime manner possible and to the lack of adequate manpower in the distributing branch of the petroleum industry.

Accordingly, Mr. Davies added, fuel oil suppliers will not find it possible to fill the orders of all their customers immediately and simultaneously. Instead it will be necessary for them to space their deliveries over a period of several weeks in order that the most effective use can be made of automotive equipment and of the limited manpower available.

Mr. Davies emphasized, however, the importance of consumers placing their orders and turning their fuel oil ration coupons over to their suppliers as soon as they obtain them from their ration boards.

This, he said, will enable each supplier accurately to estimate his fuel oil demand early and to program his deliveries to consumer storage in an orderly and progressive fashion prior to the opening of the heating season.

* * *

It is good business, as well as your patriotic duty, to save fuel—coal, oil, or gas—by heating efficiently. Uncle Sam wants you to fight waste!

Answering the Question:

How Much of My Fuel Should I Have Used?

IN THESE days of rationed fuel oil, and of an awareness generally of the need of using natural resources as sparingly as possible, data which will enable one to judge whether fuel is being consumed economically or not are of considerable help. For this purpose, Tables 1 and 2 have been prepared.

Although the illustrative examples below deal exclusively with fuel oil, these tables can be used for any kind of fuel—oil, gas, or coal—and applied to any building which is normally heated to 70 F. The reason that the tables cannot be applied to buildings which are heated to lower temperatures is, of course, that the degree day data from which these tables have been computed are based on a 70 F indoor temperature.

Table 1 shows for various localities the fuel burned in each month as a percentage of the total fuel burned during a heating season of a normal number of degree days. The values of monthly degree days and the values of normal degree days for the entire heating season are taken from pp. 232-233 of the ASHVE's Heating, Ventilating, Air Conditioning Guide, 1943.

The following example illustrates the use of Table 1.

Example 1: Suppose that the fuel oil burned in an average year by a building in Los Angeles amounts to 3000 gal. How much oil would normally be burned in the month of January?

Solution: Referring to Table 1, in Los Angeles the fuel burned during January is 21.9 per cent of the total. Hence, the fuel burned during January = 3000 × 0.219 = 657 gal.

Table 2 shows the total fuel burned from October 1 to the end of any month as a percentage of the total fuel burned during a heating season of a normal number of degree days, as October 1 is the start of the OPA's "heating year." The following example illustrates the use of Table 2.

Example 2: Take the conditions of example 1 and find the total fuel normally burned from October 1 to the end of January.

Solution: Referring to Table 2 for Los Angeles and for the period from October 1 to January 31, 48.6 per cent of the total fuel would be burned. Hence, 0.486 × 3000 = 1458 gal.

Table 1—Percentage of total fuel burned in each month for a year of normal degree days

	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	July	Aug.	Sept.
ALABAMA												
Birmingham	2.1	13.8	23.9	25.6	19.8	12.4	2.4	...	...	...	...	...
Mobile	0.2	13.0	27.0	28.4	19.6	11.1	0.7	...	...	...	...	...
ARIZONA												
Phoenix	...	11.4	28.7	30.4	19.7	9.5	0.3	...	...	...	...	...
ARKANSAS												
Fort Smith	2.9	13.5	22.8	25.4	20.0	12.3	3.1	...	...	...	...	...
Little Rock	2.5	13.5	22.5	25.6	19.7	13.0	3.2	...	...	...	...	...
CALIFORNIA												
Los Angeles	0.7	8.3	17.7	21.9	18.1	15.8	11.4	5.9	0.2	...	...	...
San Francisco	4.3	8.1	13.1	14.4	11.0	10.3	9.3	7.8	6.0	6.2	5.7	3.8
COLORADO												
Denver	7.3	12.8	17.2	18.5	15.4	13.5	9.1	4.6	0.3	...	...	1.3
Grand Junction	6.7	13.6	20.4	22.4	15.8	11.7	6.7	2.2	...	...	...	0.5
CONNECTICUT												
New Haven	5.9	11.7	17.0	19.3	17.0	15.3	9.0	3.7	0.2	...	...	0.9
DIST. OF COLUMBIA												
Washington	5.1	12.8	19.0	21.2	18.0	15.0	7.5	1.3	...	...	...	0.1
FLORIDA												
Jacksonville	...	9.5	29.1	32.1	21.1	8.2	...	...	...	...	...	...
GEORGIA												
Atlanta	2.8	13.5	21.9	24.2	19.3	14.1	4.2	...	...	...	...	...
Savannah	0.4	12.8	25.7	27.7	20.2	12.2	1.0	...	...	...	...	...
IDAHO												
Boise	7.7	12.8	18.2	19.4	15.1	12.3	7.8	4.4	0.5	...	...	1.8
ILLINOIS												
Chicago	5.1	11.8	18.0	20.5	17.5	14.8	8.6	3.3	0.1	...	...	0.3
Springfield	5.3	12.7	19.1	22.1	18.4	14.2	6.8	1.3	...	...	...	0.1
INDIANA												
Evansville	4.1	13.1	20.4	23.1	19.0	14.0	5.9	0.4	...	...	...	...
Indianapolis	5.4	12.8	19.1	21.3	17.8	14.6	7.3	1.5	...	...	...	0.2
IOWA												
Des Moines	5.6	12.5	18.9	21.7	18.0	14.1	7.0	1.8	...	...	...	0.4
Sioux City	6.2	12.7	18.4	20.7	17.5	14.2	7.3	2.0	...	...	...	1.9
KANSAS												
Dodge City	5.4	13.3	19.8	22.1	17.6	13.6	6.8	1.3	...	...	...	0.1
Topeka	4.9	13.0	20.0	22.7	18.7	13.6	6.0	1.0	...	...	...	0.1
KENTUCKY												
Lexington	5.1	13.2	19.7	21.6	18.0	14.4	7.0	1.0	...	...	...	...
Louisville	4.2	13.1	20.3	22.7	18.6	14.5	6.2	0.4	...	...	...	...
LOUISIANA												
New Orleans	...	10.2	28.6	33.0	21.2	7.0	...	...	...	...	...	...
Shreveport	1.2	13.8	25.1	28.4	20.1	10.6	0.8	...	...	...	...	...
MAINE												
Eastport	6.4	10.0	14.2	16.3	14.4	13.2	9.2	6.3	3.5	1.7	1.5	3.3
Portland	6.5	11.2	16.1	18.3	16.0	14.3	9.2	5.0	1.1	...	0.1	2.2
MARYLAND												
Baltimore	4.7	12.4	19.0	21.4	18.3	15.6	7.6	1.0	...	...	...	...
MASSACHUSETTS												
Boston	5.9	11.5	16.8	19.1	16.9	15.2	9.3	4.1	0.2	...	...	1.0
MICHIGAN												
Detroit	6.0	11.9	17.1	19.5	17.2	15.2	8.7	3.4	0.1	...	...	0.3
Marquette	6.5	10.9	15.1	17.3	15.6	14.3	9.4	5.7	2.1	0.1	0.4	2.6
MINNESOTA												
Duluth	6.6	10.7	15.5	18.1	15.3	13.1	8.6	5.6	2.4	0.3	0.8	3.0
Minneapolis	6.3	12.4	17.9	20.6	17.4	13.9	7.1	2.9	0.1	...	...	1.4
MISSISSIPPI												
Vicksburg	1.2	13.6	25.1	28.2	20.0	10.9	1.0	...	...	...	...	...
MISSOURI												
Kansas City	4.5	12.8	20.2	22.8	18.9	13.8	6.1	0.9	...	...	...	...
St. Louis	4.2	13.0	20.6	23.1	18.6	14.3	5.9	0.3	...	...	...	...
Springfield	4.8	13.1	20.2	22.1	18.9	13.9	6.1	0.9	...	...	...	...
MONTANA												
Havre	7.4	11.7	16.0	18.7	16.7	13.6	7.4	4.2	1.1	...	0.2	3.0
NEBRASKA												
Lincoln	5.5	12.8	19.2	21.6	18.0	14.1	6.7	1.8	...	...	...	0.3
Omaha	5.4	12.9	19.5	21.7	18.0	14.1	6.7	1.5	...	...	...	0.2
NEVADA												
Winnemucca	8.2	12.6	17.1	17.8	13.9	12.3	8.7	5.4	1.2	...	...	2.8
NEW HAMPSHIRE												
Concord	6.5	11.2	16.2	18.5	16.2	14.6	8.9	4.6	0.9	...	...	2.4
NEW JERSEY												
Atlantic City	4.8	11.2	17.2	19.5	17.0	15.8	10.0	4.1	0.2	...	...	0.2
NEW MEXICO												
Santa Fe	7.4	12.9	17.5	18.4	14.7	12.9	9.0	4.7	0.5	...	...	2.9
NEW YORK												
Albany	6.1	11.8	17.3	19.9	17.5	15.3	8.4	2.7	...	...	...	1.0
Buffalo	5.9	11.3	16.0	18.4	16.7	15.4	9.8	4.7	0.6	...	...	1.2
New York	5.1	11.8	17.6	20.0	17.8	16.0	8.9	2.6	...	...	...	0.2

Table 1 (continued)—Percentage of total fuel burned in each month for a year of normal degree days

	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	July	Aug.	Sept.
NORTH CAROLINA												
Raleigh	3.1	13.2	21.5	23.3	19.2	14.4	5.3	...	...	...	...	...
Wilmington	1.3	11.7	21.4	24.9	20.8	15.8	4.1	...	...	...	...	...
NORTH DAKOTA												
Bismarck	6.8	12.0	17.1	19.4	16.8	13.9	7.5	3.6	0.6	...	...	2.3
OHIO												
Cincinnati	5.6	13.2	19.1	21.0	17.6	14.5	7.4	1.4	...	...	...	0.2
Cleveland	5.7	11.8	17.0	19.4	17.1	15.3	9.2	3.6	0.1	...	...	0.8
Columbus	5.6	12.8	18.7	20.8	17.7	14.8	7.6	1.7	...	...	...	0.3
OKLAHOMA												
Oklahoma City	3.3	13.4	22.0	24.5	19.6	12.8	4.3	0.1	...	...	...	...
OREGON												
Baker	7.9	12.1	16.2	17.2	14.0	11.7	8.2	5.7	2.7	0.2	0.4	3.7
Portland	7.5	12.3	16.6	17.9	14.4	12.6	8.9	5.7	1.8	...	...	2.3
PENNSYLVANIA												
Philadelphia	4.7	12.1	18.6	21.0	18.2	15.6	8.1	1.6	...	...	...	0.1
Pittsburgh	5.6	12.6	18.4	20.5	17.7	15.2	8.0	1.7	...	...	...	0.3
SOUTH CAROLINA												
Charleston	0.5	12.0	23.9	27.2	20.5	15.7	2.2	...	...	...	...	...
Columbia	2.3	14.0	23.3	24.9	19.9	12.9	2.7	...	...	...	...	...
SOUTH DAKOTA												
Huron	6.6	12.3	17.5	20.4	17.4	13.6	7.3	3.3	0.2	...	...	1.4
Rapid City	7.1	12.1	16.4	18.5	16.1	13.9	8.6	4.7	0.7	...	...	1.9
TENNESSEE												
Knoxville	4.4	14.2	21.1	22.4	17.9	14.0	5.8	0.2	...	...	...	...
Memphis	2.6	13.5	22.4	25.3	19.6	13.3	3.3	...	...	...	...	...
Nashville	3.7	13.7	21.3	23.4	18.7	14.0	5.1	0.1	...	...	...	...
TEXAS												
El Paso	2.9	14.9	25.2	25.0	18.1	11.5	2.4	...	...	...	...	...
Fort Worth	1.1	13.1	24.9	27.9	21.5	10.4	1.1	...	...	...	...	...
Houston	...	10.7	28.8	33.2	22.3	4.9	...	...	...	...	...	...
San Antonio	...	11.3	28.7	32.3	22.0	5.7	...	...	...	...	...	...
UTAH												
Modena	8.0	13.0	16.8	18.0	14.4	12.6	8.6	5.4	0.9	...	...	2.3
Salt Lake City	6.9	12.8	18.3	19.8	15.6	12.9	8.3	4.2	0.2	...	...	1.9
VERMONT												
Burlington	6.5	11.5	16.8	19.0	17.0	14.8	8.7	3.5	0.3	...	...	1.9
VIRGINIA												
Lynchburg	5.2	13.9	20.5	22.1	17.9	14.2	6.0	0.2	...	...	...	...
Norfolk	2.7	12.2	20.3	22.6	18.7	15.6	7.3	0.6	...	...	...	...
Richmond	4.4	13.1	20.5	22.0	18.6	14.4	6.6	0.4	...	...	...	...
WASHINGTON												
Seattle	8.3	11.4	14.1	15.5	13.1	12.2	9.1	6.4	3.5	1.2	1.2	4.0
Spokane	8.2	12.6	17.0	18.4	15.0	12.4	7.9	4.6	1.1	...	...	2.8
WEST VIRGINIA												
Elkins	7.0	13.1	17.7	19.0	16.5	13.7	8.6	3.2	...	...	...	1.2
Parkersburg	5.7	13.3	19.2	21.0	17.9	14.3	7.2	1.2	...	...	...	0.2
WISCONSIN												
Green Bay	6.5	11.8	16.7	19.4	16.9	14.3	8.2	4.0	0.4	...	...	1.8
La Crosse	6.2	12.3	18.1	20.7	17.6	14.2	7.3	2.4	...	...	...	1.2
Milwaukee	6.0	11.6	16.9	19.3	16.5	14.3	8.9	4.7	0.7	...	...	1.1
WYOMING												
Cheyenne	8.3	12.1	15.1	16.3	14.1	13.2	9.6	6.1	1.8	...	0.2	3.2
Lander	8.0	12.6	16.7	17.5	14.4	12.2	8.2	5.2	1.6	...	0.2	3.4

Table 2—Accumulated percentages from October 1 to end of month. These accumulated percentages show total fuel burned from October 1 to end of any month for a year of normal degree days

	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	July	Aug.	Sept.
ALABAMA												
Birmingham	2.1	15.9	39.8	65.4	85.2	97.6	100.0	...	...	...	...	...
Mobile	0.2	13.2	40.2	68.6	88.2	99.3	100.0	...	...	...	...	...
ARIZONA												
Phoenix	...	11.4	40.1	70.5	90.2	99.7	100.0	...	...	...	...	...
ARKANSAS												
Fort Smith	2.9	16.4	39.2	64.6	84.6	96.9	100.0	...	...	...	...	...
Little Rock	2.5	16.0	38.5	64.1	83.8	96.8	100.0	...	...	...	...	...
CALIFORNIA												
Los Angeles	0.7	9.0	26.7	48.6	66.7	82.5	93.9	99.8	100.0	...	...	...
San Francisco	4.3	12.4	25.5	39.9	50.9	61.2	70.5	78.3	84.3	90.5	96.2	100.0
COLORADO												
Denver	7.3	20.1	37.3	55.8	71.2	84.7	93.8	98.4	98.7	...	...	100.0
Grand Junction	6.7	20.3	40.7	63.1	78.9	90.6	97.3	99.5	...	...	...	100.0
CONNECTICUT												
New Haven	5.9	17.6	34.6	53.9	70.9	86.2	95.2	98.9	99.1	...	...	100.0
DIST. OF COLUMBIA												
Washington	5.1	17.9	36.9	58.1	76.1	91.1	98.6	99.9	...	...	...	100.0
FLORIDA												
Jacksonville	...	9.5	38.6	70.7	91.8	100.0	...	...	...	...	...	...
GEORGIA												
Atlanta	2.8	16.3	38.2	62.4	81.7	95.8	100.0	...	...	...	...	...
Savannah	0.4	13.2	38.9	66.6	86.8	99.0	100.0	...	...	...	...	...
IDAHO												
Boise	7.7	20.5	38.7	58.1	73.2	85.5	93.3	97.6	98.2	...	...	100.0
ILLINOIS												
Chicago	5.1	16.9	34.9	55.4	72.9	87.7	96.3	99.6	99.7	...	...	100.0
Springfield	5.3	18.0	37.1	59.2	77.6	91.8	98.6	99.9	...	...	...	100.0
INDIANA												
Evansville	4.1	17.2	37.6	60.7	79.7	93.7	99.6	100.0	...	...	...	...
Indianapolis	5.4	18.2	37.3	58.6	76.4	91.0	98.3	99.8	...	...	...	100.0
IOWA												
Des Moines	5.6	18.1	37.0	58.7	76.7	90.8	97.8	99.6	...	...	...	100.0
Sioux City	6.2	18.9	37.3	58.0	75.5	89.7	97.0	99.0	...	...	...	100.0
KANSAS												
Dodge City	5.4	18.7	38.5	60.6	78.2	91.8	98.6	99.9	...	...	...	100.0
Topeka	4.9	17.9	37.9	60.6	79.3	92.9	98.9	99.9	...	...	...	100.0

These tables are based on data obtained from Weather Bureau records for a number of years. However, it is quite possible in any locality for the degree days in any one year to be greater or lower than the normal. In addition, because some months may be warmer or colder than normal, it is possible for the monthly degree days for any one year to differ somewhat from the normal degree days for that month. Nevertheless, the percentages given in Tables 1 and 2 can be of considerable help in roughly estimating whether any particular building is burning more or less fuel than it should according to the total alotted to it under the rationing regulations.

RECENT AMENDMENTS TO FUEL OIL RATIONING ORDER

An index to amendments Nos. 1 to 50 to the fuel oil rationing order appeared on pp. 171-172 of the April HPAC. Amendments 51-60 were covered on p. 233 of the May HPAC, and amendments 61-65 on p. 52, front section, June HPAC. The following information on amendments 66-71 has been taken from OPA fuel oil industry letters issued last month:

Amendment 66, effective June 10, provides that any person who is eligible to purchase a new oil fired space heater may receive a fuel oil ration if he purchases instead a used heater. Formerly he was eligible for a fuel oil ration only if he purchased a rationed new heater.

Amendment 67, effective July 1, makes provision for putting the fuel oil program under ration banking. Generally, it specifies who are required and who are permitted to open and use ration bank accounts, the number of ration bank accounts that may be opened by each depositor, the evidences which may be deposited, and the circumstances under which ration checks may be issued and accepted.

Amendment 68, effective July 1, provides for incidental changes throughout the regulations which are required in order to bring them into line with the new procedure for the issuance of renewal heat and hot water rations for the 1943-44 heating year. Among a number of such changes it provides, in the interest of simplicity, that the use of validating stamps and the writing in of serial numbers upon coupons be eliminated. It dispenses with the requirement that consumers return coupon sheet stubs and delivery receipt stubs to local boards. It requires dealers to preserve for two years, instead of one year, records of the base period deliveries to customers.

Amendment 69, effective July 1, provides for the issuance of renewal

Table 2 (continued)—Accumulated percentages from October 1 to end of month. These accumulated percentages show total fuel burned from October 1 to end of any month for a year of normal degree days

	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	July	Aug.	Sept.
KENTUCKY												
Lexington	5.1	18.3	38.0	59.6	77.6	92.0	99.0	99.9	...	...	...	100.0
Louisville	4.2	17.3	37.6	60.3	78.9	93.4	99.6	100.0	...	...	...	...
LOUISIANA												
New Orleans	...	10.2	38.8	71.8	93.0	100.0	...	...	...	...	...	...
Shreveport	1.2	15.0	40.1	68.5	88.6	99.2	100.0	...	...	...	...	...
MAINE												
Eastport	6.4	16.4	30.6	46.9	61.3	74.5	83.7	90.0	93.5	95.2	96.7	100.0
Portland	6.5	17.7	33.8	52.1	68.1	82.4	91.6	96.6	97.7	...	97.8	100.0
MARYLAND												
Baltimore	4.7	17.1	36.1	57.5	75.8	91.4	99.0	99.9	...	...	...	100.0
MASSACHUSETTS												
Boston	5.9	17.4	34.2	53.3	70.2	85.4	94.7	98.8	99.0	...	...	100.0
MICHIGAN												
Detroit	6.0	17.9	35.0	54.5	71.7	86.9	95.6	99.0	99.1	...	...	100.0
Marquette	6.5	17.4	32.5	49.8	65.4	79.7	89.1	94.8	96.9	97.0	97.4	100.0
MINNESOTA												
Duluth	6.6	17.3	32.8	50.9	66.2	79.3	87.9	93.5	95.9	96.2	97.0	100.0
Minneapolis	6.3	18.7	36.6	57.2	74.6	88.5	95.6	98.5	98.6	...	...	100.0
MISSISSIPPI												
Vicksburg	1.2	14.8	39.9	68.1	88.1	99.0	100.0	...	...	...	...	...
MISSOURI												
Kansas City	4.5	17.3	37.5	60.3	79.2	93.0	99.1	100.0	...	...	...	...
St. Louis	4.2	17.2	37.8	60.9	79.5	93.8	99.7	100.0	...	...	...	...
Springfield	4.8	17.9	38.1	60.2	79.1	93.0	99.1	100.0	...	...	...	...
MONTANA												
Havre	7.4	19.1	35.1	53.8	70.5	84.1	91.5	95.7	96.8	...	97.0	100.0
NEBRASKA												
Lincoln	5.5	18.3	37.5	59.1	77.1	91.2	97.9	99.7	...	...	...	100.0
Omaha	5.4	18.3	37.8	59.5	77.5	91.6	98.3	99.8	...	...	...	100.0
NEVADA												
Winnemucca	8.2	20.8	37.9	55.7	69.6	81.9	90.6	96.0	97.2	...	...	100.0
NEW HAMPSHIRE												
Concord	6.5	17.7	33.9	52.4	68.6	83.2	92.1	96.7	97.6	...	...	100.0
NEW JERSEY												
Atlantic City	4.8	16.0	33.2	52.7	69.7	85.5	95.5	99.6	99.8	...	...	100.0
NEW MEXICO												
Santa Fe	7.4	20.3	37.8	56.2	70.9	83.8	92.8	97.5	98.0	...	...	100.0
NEW YORK												
Albany	6.1	17.9	35.2	55.1	72.6	87.9	96.3	99.0	...	...	...	100.0
Buffalo	5.9	17.2	33.2	51.6	68.3	83.7	93.5	98.2	98.8	...	...	100.0
New York	5.1	16.9	34.5	54.5	72.3	88.3	97.2	99.8	...	...	...	100.0
NORTH CAROLINA												
Raleigh	3.1	16.3	37.8	61.1	80.3	94.7	100.0	...	...	...	...	...
Wilmington	1.3	13.0	34.4	59.3	80.1	95.9	100.0	...	...	...	...	...
NORTH DAKOTA												
Bismarck	6.8	18.8	35.9	55.3	72.1	86.0	93.5	97.1	97.7	...	...	100.0
OHIO												
Cincinnati	5.6	18.8	37.9	58.9	76.5	91.0	98.4	99.8	...	...	...	100.0
Cleveland	5.7	17.5	34.5	53.9	71.0	86.3	95.5	99.1	99.2	...	...	100.0
Columbus	5.6	18.4	37.1	57.9	75.6	90.4	98.0	99.7	...	...	...	100.0
OKLAHOMA												
Oklahoma City	3.3	16.7	38.7	63.2	82.8	95.6	99.9	100.0	...	...	...	...
OREGON												
Baker	7.9	20.0	36.2	53.4	67.4	79.1	87.3	93.0	95.7	95.9	96.3	100.0
Portland	7.5	19.8	36.4	54.3	68.7	81.3	90.2	95.9	97.7	...	...	100.0
PENNSYLVANIA												
Philadelphia	4.7	16.8	35.4	56.4	74.6	90.2	98.3	99.9	...	...	...	100.0
Pittsburgh	5.6	18.2	36.6	57.1	74.8	90.0	98.0	99.7	...	...	...	100.0
SOUTH CAROLINA												
Charleston	0.5	12.5	36.4	63.6	84.1	97.8	100.0	...	...	...	...	...
Columbia	2.3	16.3	39.6	64.5	84.4	97.3	100.0	...	...	...	...	...
SOUTH DAKOTA												
Huron	6.6	18.9	36.4	56.8	74.2	87.8	95.1	98.4	98.6	...	..	100.0
Rapid City	7.1	19.2	35.6	54.1	70.2	84.1	92.7	97.4	98.1	...	...	100.0
TENNESSEE												
Knoxville	4.4	18.6	39.7	62.1	80.0	94.0	99.8	100.0	...	...	...	...
Memphis	2.6	16.1	38.5	63.8	83.4	96.7	100.0	...	...	...	...	...
Nashville	3.7	17.4	38.7	62.1	80.8	94.8	99.9	100.0	...	...	...	...
TEXAS												
El Paso	2.9	17.8	43.0	68.0	86.1	97.6	100.0	...	...	...	...	...
Fort Worth	1.1	14.2	39.1	67.0	88.5	98.9	100.0	...	...	...	...	...
Houston	...	10.7	39.5	72.8	95.1	100.0	...	...	...	...	...	...
San Antonio	...	11.3	40.0	72.3	94.3	100.0	...	...	...	...	...	...
UTAH												
Modena	8.0	21.0	37.8	55.8	70.2	82.8	91.4	96.8	97.7	...	...	100.0
Salt Lake City	6.9	19.7	38.0	57.8	73.4	86.3	94.6	98.8	99.0	...	...	100.0
VERMONT												
Burlington	6.5	18.0	34.8	53.8	70.8	85.6	94.3	97.8	98.1	...	...	100.0
VIRGINIA												
Lynchburg	5.2	19.1	39.6	61.7	79.6	93.8	99.8	100.0	...	...	...	...
Norfolk	2.7	14.9	35.2	57.8	76.5	92.1	99.4	100.0	...	...	...	...
Richmond	4.4	17.5	38.0	60.0	78.6	93.0	99.6	100.0	...	...	...	...
WASHINGTON												
Seattle	8.3	19.7	33.8	49.3	62.4	74.6	83.7	90.1	93.6	94.8	96.0	100.0
Spokane	8.2	20.8	37.8	56.2	71.2	83.6	91.5	96.1	97.2	...	...	100.0
WEST VIRGINIA												
Elkins	7.0	20.1	37.8	56.8	73.3	87.0	95.6	98.8	...	...	...	100.0
Parkersburg	5.7	19.0	38.2	59.2	77.1	91.4	98.6	99.8	...	...	...	100.0
WISCONSIN												
Green Bay	6.5	18.3	35.0	54.4	71.3	85.6	93.8	97.8	98.2	...	...	100.0
La Crosse	6.2	18.5	36.6	57.3	74.9	89.1	96.4	98.8	...	...	...	100.0
Milwaukee	6.0	17.6	34.5	53.8	70.3	84.6	93.5	98.2	98.9	...	...	100.0
WYOMING												
Cheyenne	8.3	20.4	35.5	51.8	65.9	79.1	88.7	94.8	96.6	96.8	...	100.0
Lander	8.0	20.6	37.3	54.8	69.2	81.4	89.6	94.8	96.4	...	96.6	100.0

heat and hot water rations for the 1943-1944 heating season. It sanctions the use of the simple reapplication form OPA R-1167. Provision is made for raising the area ceiling from 2000 to 3000 sq ft for the first occupant in a private dwelling, and to re-figure rations for both central heating plants and space heaters used in smaller homes, as outlined in industry letter No. 11. It provides for the issuance of heating rations with class 3, 4, 5, and 6 coupon sheets, or fuel oil deposit certificates, depending on the size of the ration. The number of zones in the limitation area is increased from four to 10 by using north-south division lines running across the existing thermal zones.

Amendment 70, effective July 14, amends the definition of additional facilities to permit the issuance of fuel oil rations for the operation of incubators, brooders, growing batteries and other farm equipment, regardless of the date of its installation or acquisition. It is essential to the war effort that rations should be issued for oil burners, heaters, and other equipment for raising and preparing for market crops, poultry, livestock, or other agricultural products.

Amendment 71, effective July 7, provides for certain changes in substance and procedure in the sections of the regulations relating to control and audit. The classification of secondary suppliers is eliminated and all distributors who do not qualify as primary suppliers are defined as "dealers". All distributors registered as secondary suppliers prior to July 1, 1943, are automatically registered as dealers. Provision is made for late registration of primary suppliers and dealers. The provisions regarding commingling are changed to treat dealers and primary suppliers separately. Sections describing fuel oil inventory and storage capacity are revised for clarification. Provision is made to permit surrender of coupons in the industry within 15 days after transfer of fuel oil, with the consent of the transferor, instead of five days as presently established. Delivery of coupons in advance of the transfer of oil is limited to five days and the use of coupons delivered in advance is prohibited until the oil is actually transferred. Provision is made for the issuance of exchange certificates by boards instead of inventory coupons to dealers, after July 1. A requirement that all exchange certificates be endorsed on the back is provided, as well as the endorsement and filling in of complete information on gummed sheets. OPA R-120.

Amendment 6 to Supplement 1, effective July 1, establishes the value of the unit value coupons on class 4, 5, and 6 coupon sheets at 10 gal, 50 gal, and 250 gal, respectively, for period 1.

MEASURING the FLOW of FLUIDS

William Goodman Gives Practical Data on the Use of Orifices and Manometers

After the pressure difference across an orifice has been measured by means of a manometer, the quantity of liquid flowing through the orifice can be computed by means of the following formula, which is obtained by multiplying by $(60 \times \pi/4)$ equation 1 on p. 8 of the American Society of Mechanical Engineers publication, Fluid Flow—1940:

$$W = 31.5\ KD^2\sqrt{pd} \quad \ldots\ldots\ldots [1]$$

The value of p in this formula is the actual pressure difference in pounds per square inch across the orifice. The method of computing this pressure difference from the reading of the manometer is discussed in a later section.

The flow coefficient K of the orifice depends upon Reynolds number. Reynolds number is a composite number computed from the viscosity, density, and velocity of the fluid, and the diameter of the orifice. Regardless of the kind or form of the fluid—whether liquid, vapor, or gas—the value of the flow coefficient K depends only upon the value of Reynolds number. No matter how much the physical properties of two fluids differ, if the conditions of flow are such that the Reynolds number is the same for each of them, the flow coefficient K of the orifice will be identical for both fluids. Consequently, the importance and convenience of Reynolds numbers in computing flow coefficients is readily apparent.

Flow Coefficients

The value of K, the flow coefficient, is given for various Reynolds numbers and various size pipes in a series of charts on pp. 47 to 54 inclusive of the ASME publication, Flow Measurement—1940. However, in order to compute Reynolds number, the quantity of fluid flow-

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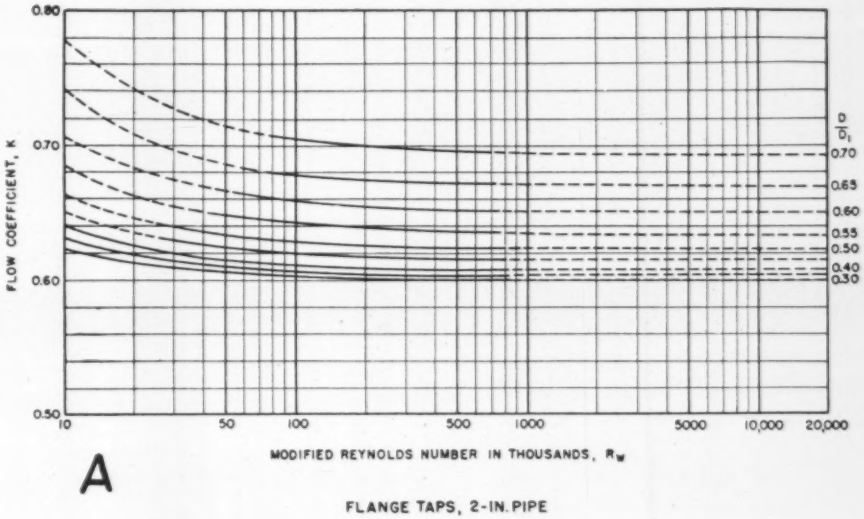
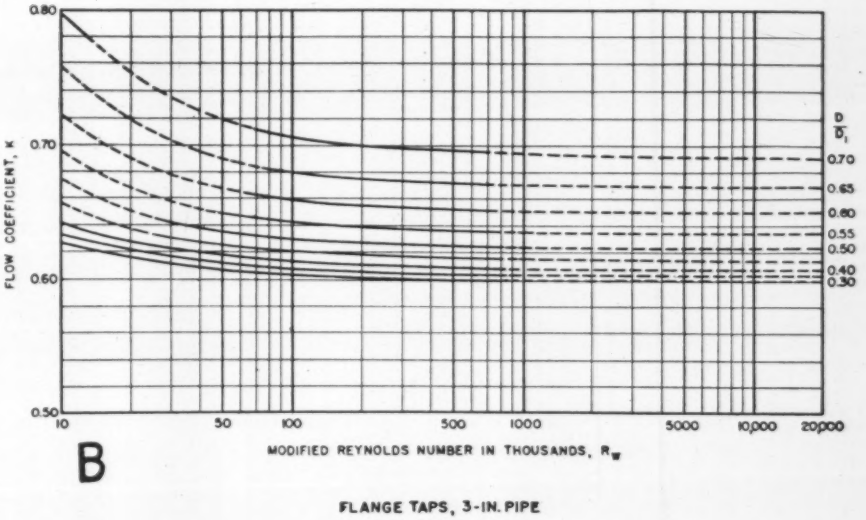
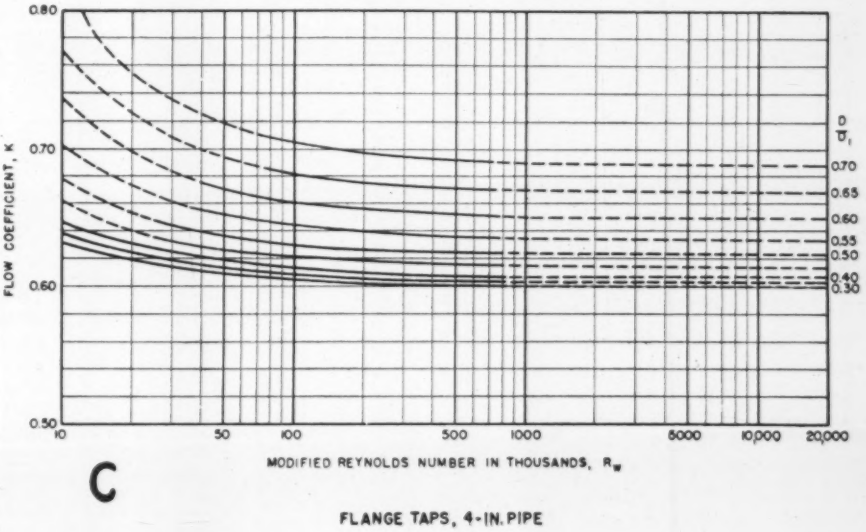
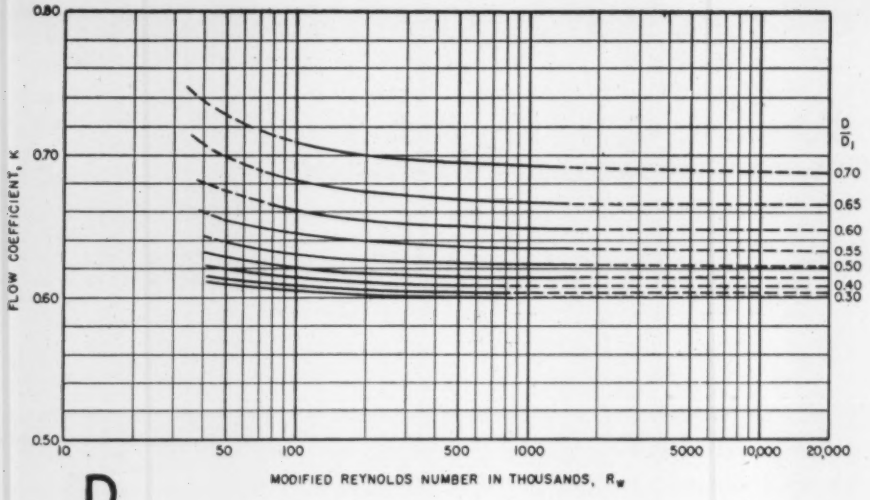
To avoid necessity of using trial and error methods when computing flow through an orifice, flow coefficients K are plotted in Charts A to S against a modified Reynolds number, R_w

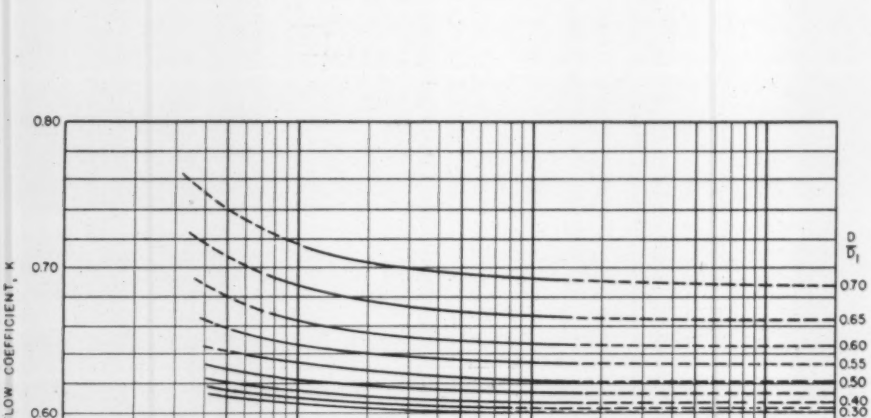




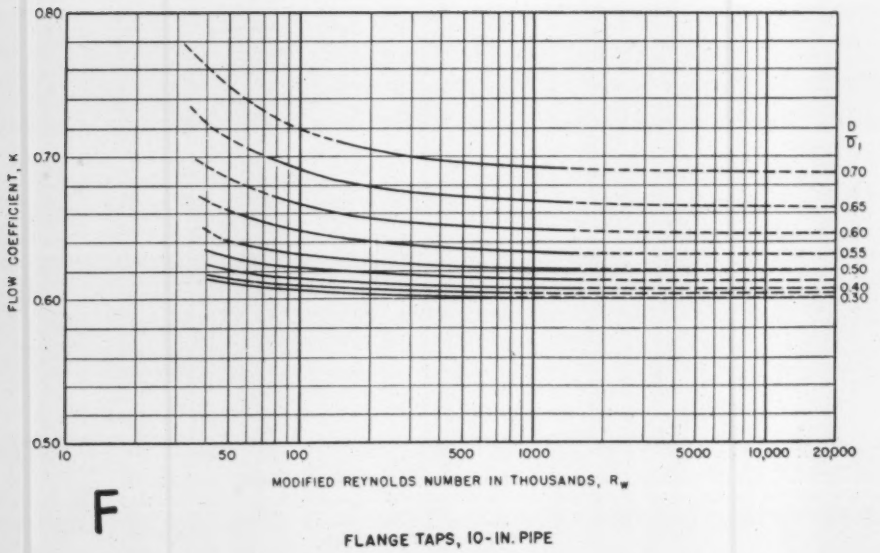




FLANGE TAPS, 6-IN. PIPE



FLANGE TAPS, 8-IN. PIPE



FLANGE TAPS, 10-IN. PIPE

ing through the orifice must first be known. In other words, before the value of K can be read from the ASME charts, the quantity of fluid flowing through the orifice must be determined. But, as is evident from equation 1 above, the quantity of fluid cannot be determined unless the value of K is known. Hence, trial and error methods are necessary when using the ASME charts to determine from a measured pressure drop the quantity of fluid flowing through an orifice.

To avoid the necessity of trial and error methods when computing the flow through an orifice, the flow coefficients K have been plotted in Charts A to S, inclusive, of this article, against a modified Reynolds number, R_w. This modified Reynolds number can be computed without knowing the weight of fluid flowing through the orifice. The only quantity that need be known is the pressure drop across the orifice as measured by the manometer. This modified Reynolds number for orifices is similar to the method—published on p. 625 of the October, 1937, issue of Heating, Piping & Air Conditioning—of using modified Reynolds numbers in connection with piping flow problems. For a further discussion of these modified Reynolds numbers for orifices, see Alternative Forms of the Reynolds Numbers, by Ed. S. Smith, Jr., p. 201, August, 1935, issue of Instruments.

The modified Reynolds number* to be used in determining the flow co-

*Reynolds number may be written (see p. 627 of October 1937, HPAC):

$$R_p = 378.8 \frac{W}{Du} \quad \text{.....[a]}$$

Eliminating W between [a] above and equation 1 of the text, the following is obtained:

$$R_p = 11{,}940\, K \frac{D}{u} \sqrt{pd} \quad \text{....[b]}$$

Let

$$R_w = 11{,}940 \frac{D}{u} \sqrt{pd} \quad \text{.....[c]}$$

This is equation 2 of the text. Substituting [c] into [b],

$$R_w = \frac{R_p}{K} \quad \text{.....[d]}$$

Corresponding values of R_p and K were read from Figs. 34a to 36d of the ASME publication, Fluid Flow—1940. (On the ASME figures, R_p is marked R_d.) The values of R_w plotted in Charts A to S inclusive of this article were then computed by means of [d] above.

To avoid necessity of using trial and error methods when computing flow through an orifice, flow coefficients *K* are plotted in Charts A to S against a modified Reynolds number, R_w

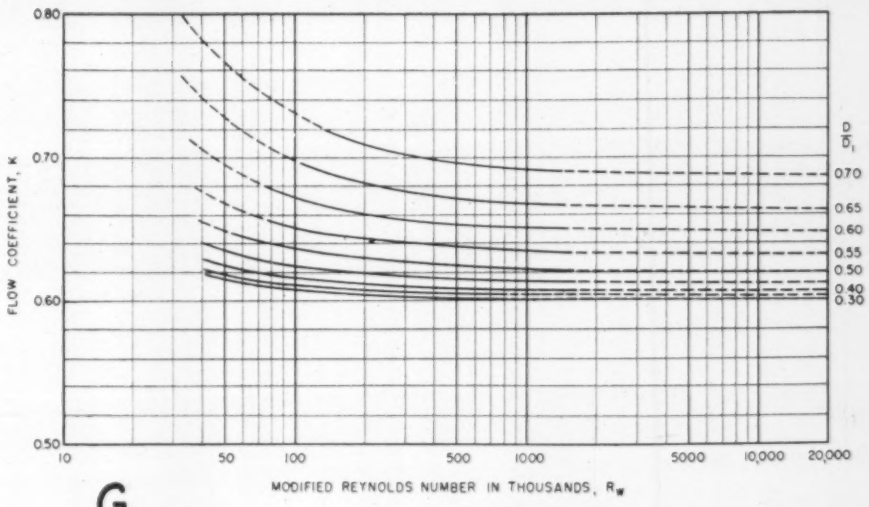
efficient K from Charts A to S inclusive is computed by means of the following formula:

$$R_w = 11{,}940 \frac{D}{u} \sqrt{pd} \quad \ldots\ldots [2]$$

In addition to the Reynolds number, the value of the flow coefficient K depends upon the ratio of the orifice diameter to the diameter of the inside of the pipe. This ratio, D/D_1, is called the *diameter ratio*. The diameter ratio is important because included in the value of K is the correction for the velocity of the fluid approaching the orifice, and this correction depends only upon the diameter ratio.

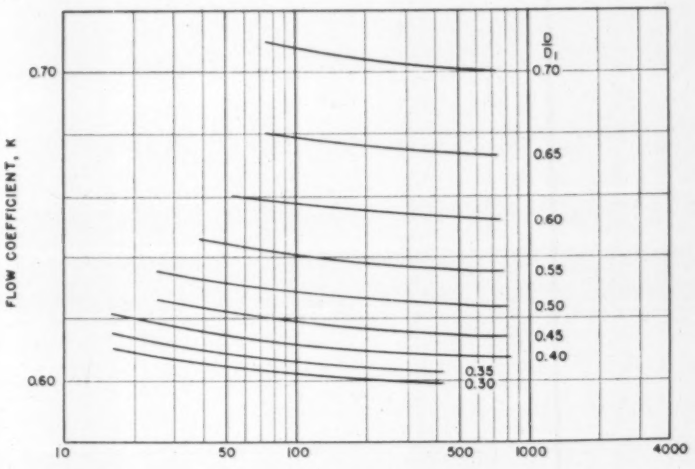
K and Pressure Tap Location

Connections must be made from the manometer to pressure taps in the pipe line on either side of the orifice. The location of these pressure taps in relation to the orifice also affects the value of the coefficient K. The values of K have been analyzed and standardized by the ASME for three different locations of the pressure taps in relation to the orifice. These three locations of the pressure taps are called



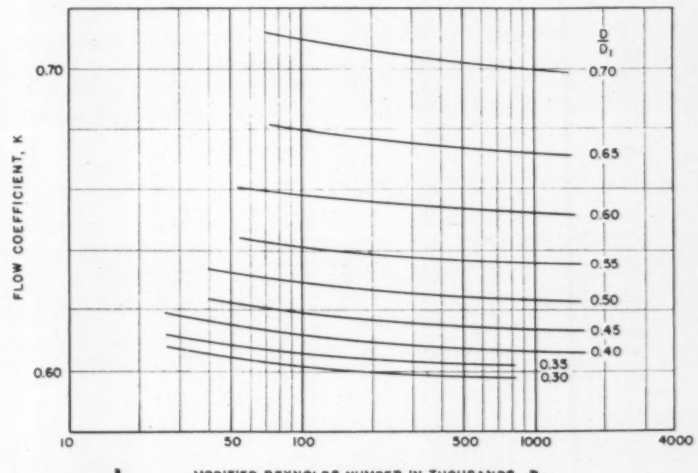
G

FLANGE TAPS, 14-IN. PIPE



H

VENA CONTRACTA TAPS, 2-IN. PIPE



J

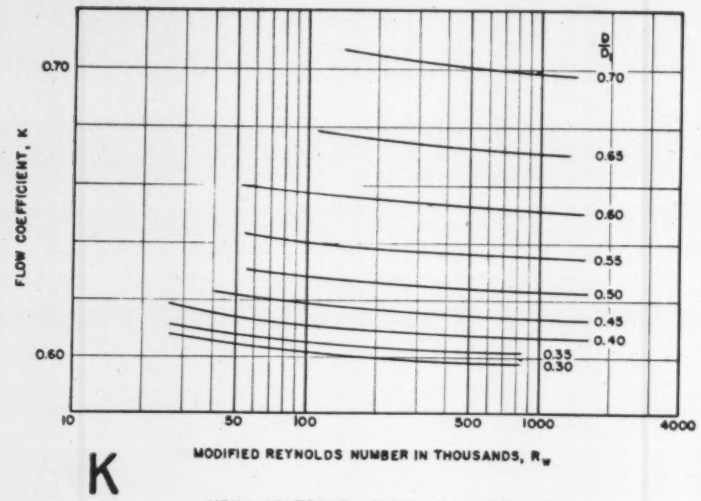
VENA CONTRACTA TAPS, 3-IN. PIPE

For Practical Men

Although orifices have been used for many years, the data required for selecting and using them has not always been generally available in a form suitable for the needs of practical men. True, codes for making measurements of fluid flow are available, but the arrangement and formalized language of such codes has not lent itself to quick and ready use by those who have occasion to select and use orifices only occasionally.

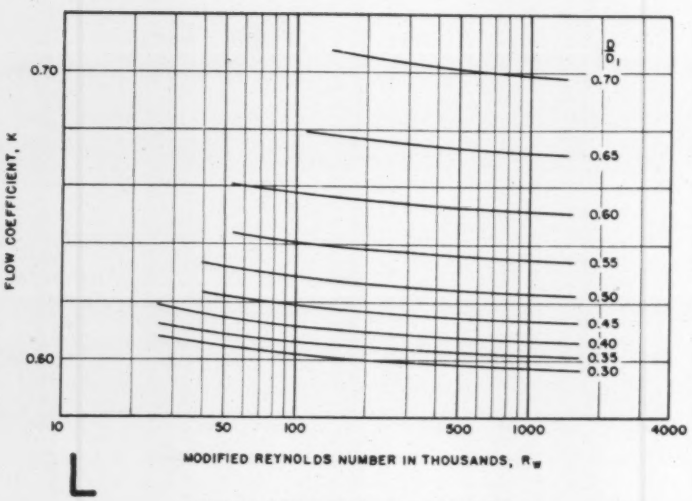
In the article of which this is the first part, the method of selecting proper combinations of orifices and manometers for measuring any type of fluid is given. In addition, this article contains new and more convenient charts for determining the flow coefficient of an orifice for any fluid whether it be water, brine, steam, air, or any other. Finally, practical pointers are given on the construction, installation, and operation of orifices and manometers for metering almost any liquid, vapor, or gas.

Mr. Goodman is consulting engineer, the Trane Co., and a member of HPAC's board of consulting and contributing editors.



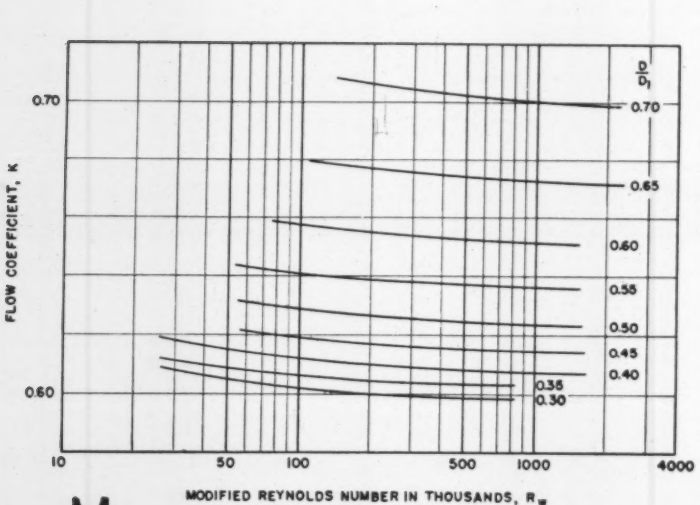
K

VENA CONTRACTA TAPS, 4-IN. PIPE



L

VENA CONTRACTA TAPS, 6-IN. PIPE



M

VENA CONTRACTA TAPS, 8-IN. PIPE

flange taps, vena contracta taps, and radius taps.

The distinction between these taps and their location in relation to the orifice plate will be discussed in detail later. For the moment it is sufficient to be aware that the flow coefficient K has been determined and standardized for these three types of taps. The value of the flow coefficient K is different for each type of tapping. Charts A to G inclusive are for flange taps for various size pipes; Charts H to O inclusive are for vena contracta taps; and Charts P to S inclusive are for radius taps.

Use of Equations

The following example illustrates the use of equations 1 and 2 for computing the quantity of fluid flowing through an orifice installed in a pipe line.

Example 1: An orifice whose diameter is 1.534 in. is installed in a 3 in. standard weight steel pipe (3.068 in. inside diameter). The density of the water is 62.3 lb per cu ft, and its viscosity is 1 centipoise. The pressure difference across the orifice is 6.67 psi and flange taps are used. Find the quantity of water flowing through the orifice.

Solution: Using equation 2:

$$R_w = 11{,}940 \times \frac{1.534}{1}\sqrt{6.67 \times 62.3}$$

$$= 373{,}300$$

Diameter ratio of orifice, $D/D_1 = 1.534/3.068 = 0.5$.

Referring to Chart B, which is drawn for 3 in. pipe with flange taps, for a modified Reynolds number R_w of 373,300 and for a diameter ratio $D/D_1 = 0.5$, the value of K read from the chart is 0.626. Using equation 1:

$$W = 31.5 \times 0.626 \times (1.534)^2\sqrt{6.67 \times 62.3}$$

$$= 945.7 \text{ lb per min of water}$$

The quantity of liquid flowing is usually expressed in gallons per minute. The following equation can be used to convert weight to gallons.

$$\text{Gpm} = 7.48 \times \frac{W}{d} \quad \ldots\ldots\ldots [3]$$

Example 2: Take the conditions of the preceding example and find the quantity of water flowing in gallons per minute.

Solution: Substituting in equation 3:

To avoid necessity of using trial and error methods when computing flow through an orifice, flow coefficients K are plotted in Charts A to S against a modified Reynolds number, R_w

Symbols

D = throat diameter of orifice in plate, in.;
D_1 = inside diameter of pipe in which orifice is installed, in.;
d = density of fluid at pressure P *ahead* of orifice, lb per cu ft;
K = flow coefficient of orifice;
p = pressure difference across orifice, psi;
R_p = Reynolds number, $\frac{4}{\pi}\frac{W}{Du}$.
R_w = modified Reynolds number for selecting flow coefficients;
u = viscosity of fluid flowing through orifice, centipoise;
W = weight of fluid flowing through orifice, lb per min.

$$\text{Gpm} = 7.48 \times \frac{945.7}{62.3}$$

= 113.5 gpm flowing through the orifice.

How to Use Charts

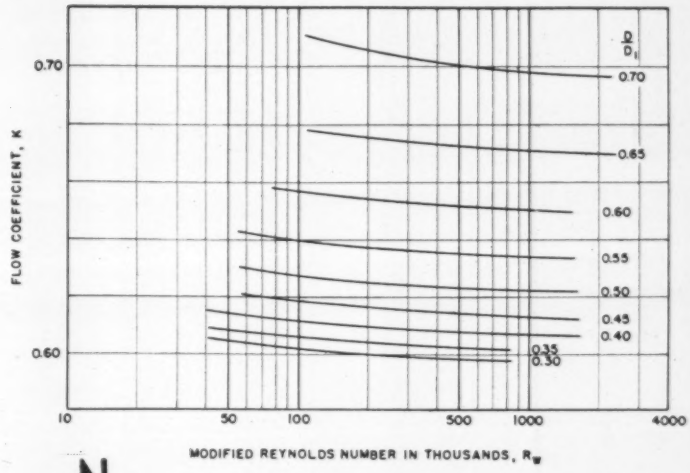
The dashed sections of the curves on some of Charts A to S represent extrapolated, not experimental, data. However, values of the flow coefficient K read from these sections of the curves are sufficiently accurate for all practical purposes.

Notice in Charts A to S that the value of the flow coefficient approaches a constant value as Reynolds number increases. For larger Reynolds numbers than are shown on the charts, use the value of K corresponding to the largest value of Reynolds number shown on a given chart.

Although charts are given for nearly all pipe sizes, occasionally it may be necessary to install an orifice in an intermediate size pipe for which no chart is given. For such an intermediate pipe size, interpolate for values of K between the two nearest pipe sizes for which charts are given.

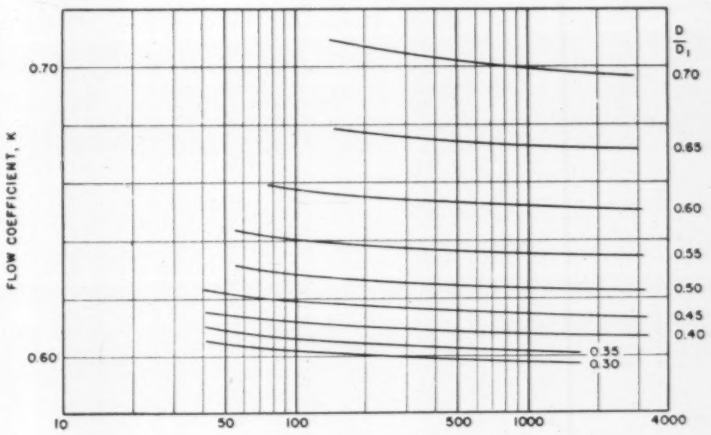
When using standard weight steel pipe, use the chart for the correct pipe size. However, for smooth pipe such as brass or copper, disregard the actual pipe size and select the flow coefficient K from the chart for the *largest* pipe size shown for the particular pressure taps used.

The viscosities of liquids and gases are practically independent of pressure, and vary only with the temperature. The viscosity of a saturated vapor such as steam



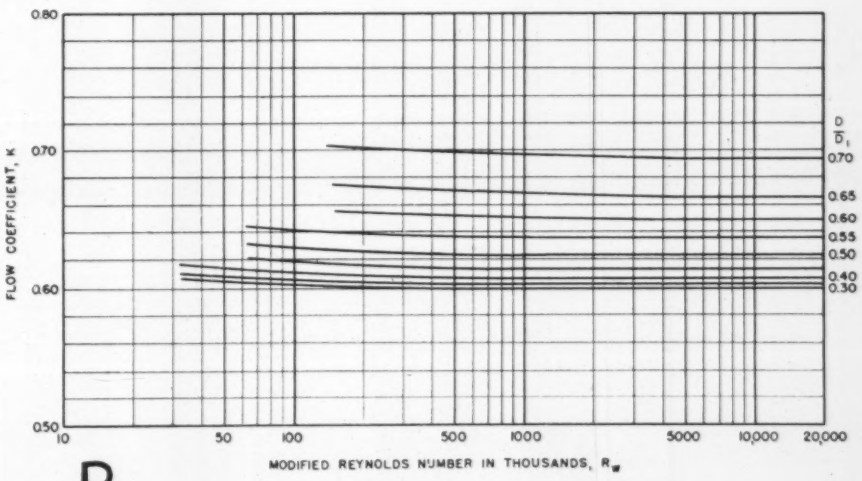
N

VENA CONTRACTA TAPS, 10-IN. PIPE



O

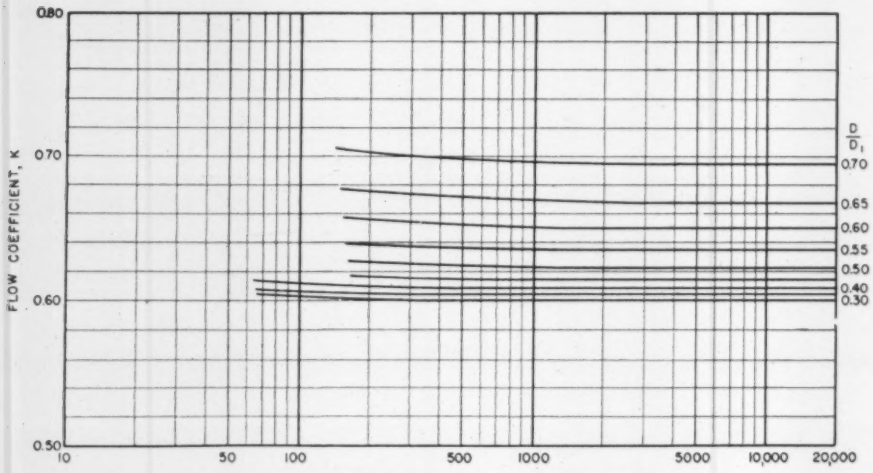
VENA CONTRACTA TAPS, 14-IN. PIPE



P

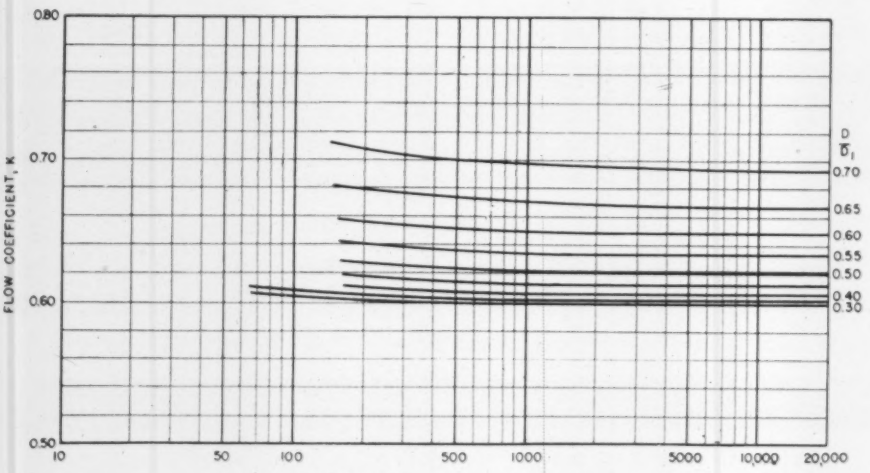
RADIUS TAPS, 2-IN. PIPE

varies with both temperature and pressure. The viscosities of many different substances can be obtained from such reference books as Perry's Chemical Engineers Handbook or the International Critical Tables. For convenience, three figures and a table showing some viscosity data will be published in a succeeding part of this article.



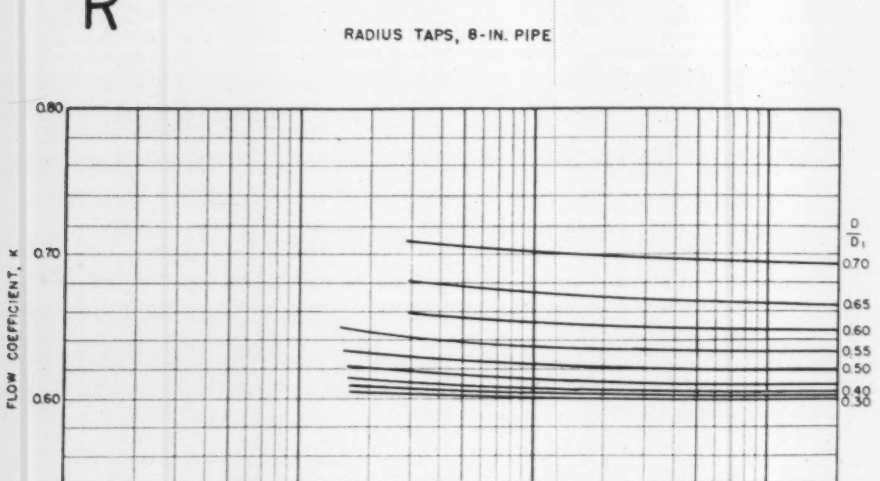
Q

RADIUS TAPS, 4-IN. PIPE



R

RADIUS TAPS, 8-IN. PIPE



S

RADIUS TAPS, 16-IN. PIPE

DESERT WARRIORS USE AIR CONDITIONING

The story of how air conditioning was called upon to help solve problems involved in desert warfare was told recently by G. E. Weiss, of Gay Engineering Corp., Carrier dealer.

During desert maneuvers in subtropic heat, with temperatures inside armored vehicles soaring to 120 to 130 deg, U. S. Army officers were faced with the problem of severe heat prostration calling for medical treatment for the troops "on the spot" in order to obviate the possibility of fatalities.

Considering the plausibility of using refrigerated trucks as "portable hospitals," it was found that such trucks were not available in less than six to eight weeks' time and a truck body building firm estimated that it would take from three to four weeks to alter available standard Army moving van trailers for the particular purpose.

As a last resort, the two available Army moving vans were brought to the Gay yard. Truck refrigeration units were installed, and the vans were back on the road in five days. The work included removal of the roof to allow for insulation; raising the walls and providing complete insulation for all sections of each of the two trailers, including walls, roof and floor; and constructing two complete insulated steel refrigeration doors for the back section. A complete truck refrigeration unit was installed in each of the vans, with the compressor suspended underneath the chassis. In addition to serving the air conditioning system, the refrigerating unit supplies chilled water to a storage tank used in treatment of heat prostration.

The duty of the refrigeration system is, first, to maintain a temperature within the trailers of not in excess of 68 F, and second, to maintain a water temperature in a tank containing 150 gal of not in excess of 40 F. In addition to the refrigerating and air conditioning system, each of the trailers was equipped with a surgical cabinet and copper sink with water connections and drain, a sheet metal tub of sufficient size to completely immerse a man, steel stretcher supports to accommodate six stretchers, and a special spotlight for use in the performance of any surgical work that might be required.

In use as portable air conditioned receiving stations in which to treat men in the field suffering from heat stroke and heat exhaustion, the units quickly proved their worth. A letter from the major general in command of the training center stated: "The workmanlike construction of the remodeling and installation of the equipment are excellent in every detail and the two units have been placed in immediate service and are rapidly proving their value."

Designing Warship Ventilation with Standardized Equipment

By Thomas H. Urdahl, Comdr., USNR, Officer-in-Charge, Air Cond. Section, Shipbuilding Div., Bureau of Ships, Navy Dept., and W. C. Whittlesey, Senior Engineer, Air Cond. Section, Shipbuilding Div., the Bureau of Ships

A PREVIOUS discussion of the merits of standardized heating and ventilating equipment for fighting ships, published in the July HPAC, only indirectly emphasized the effect of standardization upon design and the simplification of the work of the designer. Too much emphasis cannot be placed upon the fact that standardized equipment not only simplifies the designer's problem but it insures accuracy in the final installation which is not possible without exact foreknowledge of dimensions, characteristics, weights, and capacities of the equipment which will be installed, thus enhancing the exercise of engineering judgment applied to a given problem.

Too Liberal Safety Factors

The traditional method of equipment procurement for naval vessels was to specify the required characteristics to be met by the product offered by the successful bidder. Naval service usually requires specialized materials, and this seems to have fostered the belief that so long as the materials of a standard manufactured article had to be changed, performance characteristics could also be altered with little difficulty, and specifications were developed to require equipment to fit a specified design with elaborate tests of each unit to assure compliance with the minimum requirements.

Actually, manufacturers did not design special equipment to fit each naval unit, but selected designs from their own standardized line to satisfy the requirements. The specifications were usually for minimum values, and as the tests were severe the suppliers tended to become quite conservative, with the result that liberal safety factors were usually incorporated. This factor, together with the general practice of selecting equipment from the manufacturer's line, resulted in the installation of oversized equipment and the consequent undesirable increase in weight, space, and power requirements.

This type of specification and procurement relieved the designer of responsibility for equipment performance. Bids for equipment were advertised only after the basic design was completed, and the designer had no way of determining who the supplier would be or what characteristics the equipment might have. Accordingly, the design was developed with complete disregard for the equipment characteristics, with the hope that the ultimate equipment selection—usually made by the successful bidder — would somehow be satisfactory.

An attempt was made some years ago partially to standardize ventilation fans. This was done primarily for procurement reasons, as the normal method required so much time that ventilation fans often held up construction even in the comparatively leisurely prewar period. Fans of various capacities were purchased from various manufacturers for stock under the requirement that they should deliver their rated capacity at 2.0 in. WG static pressure. This resulted in all systems being designed for 2 in. static pressure loss regardless of the system length or type of service. This low pressure loss required large ducts and generous cuts in protective or strength structure for the longer systems. These requirements were reflected in the general arrangements of the vessels, and the difficulties in finding space and the reluctance to allow large structural cuts tended to minimize the quantity of ventilation, and resulted in exploiting the access arrangements to the maximum extent for natural ventilation — particularly in connection with exhaust provisions. This expedient was generally satisfactory for peacetime operation, but wartime cruising requires the access to be kept closed for compartmentation, which has necessitated extensive alterations on vessels designed in this manner.

Fan Noise Problem

At the start of the new building program in the early 30's it became evident that space allotments for ventilation would have to be reduced, and the design procedure reverted to the traditional practice of designing systems without re-

Effects Economies

In the July issue of Heating, Piping & Air Conditioning, Comdr. Urdahl and Lt. John Everetts, Jr., stressed the importance of the Navy's standardization program, giving the reasons for it and showing its application to various items of heating, ventilating, and air conditioning equipment. This month, Comdr. Urdahl and Mr. Whittlesey discuss the advantages of standardization of equipment from the designer's viewpoint, showing how it simplifies the design problem and insures accuracy of the final installation.

Factual knowledge of equipment characteristics enables the designer to effect economies that were never possible without reliable performance information and a procurement procedure that assures compliance with the established standards, the authors conclude.

gard for the available equipment. The industry supplied fans to satisfy the volume and pressure specifications, but the noise characteristics were more reminiscent of a fire siren than a ventilation blower. This noise situation became so acute that it interfered with navigation, battle problems, and the rest and sleep of personnel.

The noise problem eventually led to the development of satisfactory fans by several manufacturers, but this improvement had little effect on the design policy except to permit the use of high static pressures. If anything, this change complicated the situation inasmuch as the range of pressures in common use was extended and, consequently, each manufacturer had to expand the line from which many fans were chosen, thus increasing the number of fan sizes required for a given ship. In an effort to reduce the number of fan sizes, larger discrepancies were accepted between the required fan characteristics and the actual fan performance which tended further to unbalance the design.

Advantage of Designing for Available Equipment

Preliminary studies of the standardization problem with a view to reducing the number of fan sizes demonstrated that considerable savings would be possible if duct systems were designed to the actual fan characteristics. The advantages of designing for the available equipment can be demonstrated by a simple example. Fig. 1 shows a diagrammatic arrangement of a system for which duct sizes are to be selected. One method in common use is to design the system for 0.01 in. loss per foot of duct. This method, together with the nominal loss of 0.6 in., through both heaters, would require a 14 in. duct and a fan rated at 3000 cfm at 2.52 in. The closest available standard fan supplies 3000 cfm at 3.30 in., and standard heaters to satisfy the temperature requirements would require a total of 0.42 in. instead of 0.60 in. By designing this system to suit the standard fan (3000 at 3.30 in.), and making allowance for the actual heater loss (0.42 in.) the main duct diameter could be reduced to 13 in. This would permit a weight reduction of about 15 per cent for ducts and duct fittings (such as watertight closures).

Failure to consider equipment characteristics can also throw a system out of balance. For example, the proper standard heater for branch *A* required only 0.11 in. pressure loss, although the heater for branch *B* required 0.27 in. loss. If both of these branches were sized for the nominal heater resistance of 0.20 in., branch *A* would be 9 in. and branch *B* would be 11 in. However, because of the actual difference in heater losses, these branches would deliver about 1500 cfm each instead of the required quantities. Branch *A* should be 8½ in. and branch *B* 12 in. for proper balance, but to arrive at these sizes the equipment performance must be considered.

One design was investigated during the standardization studies that used 37 fan sizes for 153 ventilation systems. It was found that slight changes in duct design that would have resulted from consideration of fan characteristics, would reduce the number of fan sizes to eight—a potential reduction of 29 sizes. Thus, designing a system to suit the equipment characteristics not only results in better arrangements but also simplifies the procurement processes and reduces the cost by permitting mass production of superior standardized articles.

The standardization of ventilation equipment other than fans—such as air coolers, heaters, terminals, and other fittings — will show similar savings as soon as the designers alter their method of approach to consider the characteristics of the available equipment in the preliminary layout stages. This is a departure from orthodox practice for many who have always had to make layouts without knowing the equipment performance. However, regardless of the stage of development a design may be in, standardized products will enable the designer to know the performance characteristics regardless of who the supplier may be, and he can incorporate the equipment selection in his design.

Control Over Safety Factors

The selection of equipment by the designer also gives him control over the safety factors. Whenever equipment must be chosen by people unfamiliar with the basic problem, there is a conservative tendency that almost inevitably results in oversized selections. This policy is almost universally acceptable to inspectors who think in terms of minimum requirements, although there are many cases where overcapacity causes trouble.

For example, the exhaust ventilation for hot compartments is designed to be in excess of the supply in order to maintain an indraft through the access, and thus prevent hot air from entering and overheating contiguous spaces. If the supply fans are chosen with too liberal a safety factor, they may overcome the designed differential and actually deliver more air than the exhaust fans can handle, which produces an outdraft of hot air that overheats the adjacent country. (It might be expedient to add that this example is not an academic hypothesis but, unfortunately, is a

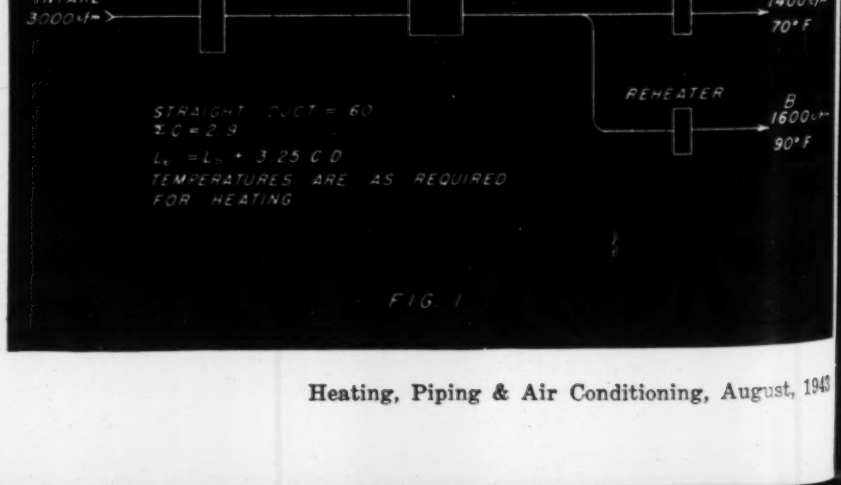


FIG. 1

On Fig. 1, C = pressure loss in equivalent diameter, L_e = total equivalent length, L_s = total length in straight pipe, D = diameter of largest pipe.

condition that is frequently encountered.) The intelligent application of standardized equipment by the designer will eliminate such conditions.

Procurement of Equipment

The procurement of equipment is also a major factor in design, and any expedient that will assure the availability of the necessary fittings is a vital aid. There is no object in developing a design — however perfect — if the necessary equipment cannot be obtained. Standardization is particularly important for repair and alteration items when time is limited and manufacturing facilities are distant. Even if equipment stocks are not available, standardization permits robbing a ship in for extended overhaul to repair minor damage on another vessel in order to return it to immediate combat duty.

Equipment standardization is not only the key to material procurement, but is also the medium by which the designer can improve his layouts and accurately control ultimate shipboard performance. The factual knowledge of equipment characteristics enables him to effect design economies that were never possible without reliable performance information and a procurement procedure that assures compliance with the established standards.

Air Conditioning a Tool of the Testing Engineer

Temperature, Humidity and Testing Speed Most Important Variables in Testing Plastics, Says William N. Findley, Associate in Theoretical and Applied Mechanics, University of Illinois

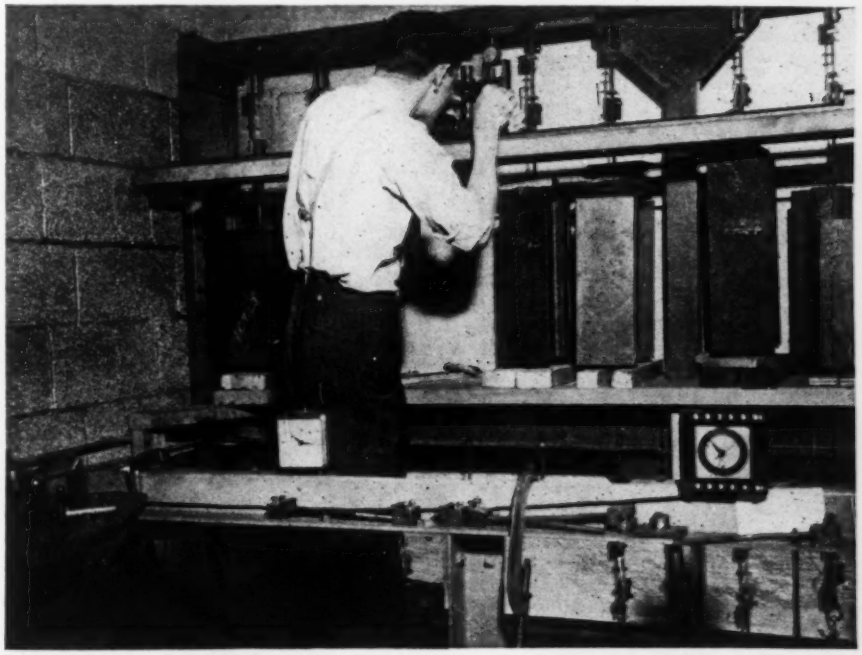
AN AIR conditioned laboratory for testing the strength of engineering materials is a relatively new requirement. With metals and many other building materials, small variations in temperature have little effect upon the results of mechanical tests and, for most materials, variations in humidity have no measureable effect. One notable exception is that of wood; the moisture content of wood must be accurately maintained if significant test results are desired. The new class of materials known commercially as plastics has brought to the materials testing engineer increased difficulties, however, because in the testing of plastics there are a number of variables which are of extreme importance. The three most important of these variables are temperature, relative humidity, and speed of testing.

Plastics Classified by Their Reaction to Heat

Plastics are commonly classified into two groups according to their reaction to heat. The first group, called thermoplastics, can be repeatedly softened on heating and hardened on cooling. The second group, thermosetting materials, undergoes a chemical change under heat and pressure which causes the material to become permanently hard and infusable. As might be expected, thermoplastic materials are much more sensitive to small changes in temperature than thermosetting materials.

The effect of temperature changes on the mechanical properties of a thermosetting material, methyl methacrylate, under both tension and compression loads has been shown by W. F. Bartoe [see Mechanical Engineering, Vol. 61, No. 12, December, 1939]. A change from 68 to 86 F resulted in about 15 per cent decrease in yield point in tension and about 17 per cent decrease in yield point in compression. The effect of changes in temperature on the mechanical properties of thermosetting materials, while less pronounced than for thermoplastics, is still important. A study of the effect of temperature on the strength properties of different phenolic molding com-

Creep testing of plastics



positions was contained in a paper by Carswell, Telfair, and Haslanger [see Modern Plastics, Vol. 19, No. 11, July, 1942]. With these materials a change from 68 to 86 F resulted in a decrease in ultimate tensile strength of about 4 per cent.

Effect of Moisture Content Changes

The effect of changes in moisture content has been studied at the University of Illinois. In one series of tests, two groups of compression specimens of cellulose acetate (a thermoplastic) were subjected to two different humidity conditions. One group was placed in a calcium chloride desiccator for 48 hr and another group was immersed in water for a like period. Following this treatment specimens were tested in compression at intervals of time for about 13 months in a room maintained at a constant temperature of 77 F and constant relative humidity of 50 per cent.

The yield point as obtained from these tests was plotted against the duration of time after the humidity treatment. It was noticed that the dried specimens were about 50 per cent stronger than the wet specimens immediately after the drying or wetting treatment, and that about 500 hr (three weeks) was required for the two groups of samples to approach equilibrium in an atmosphere maintained at a constant temperature of 77 F and a constant relative humidity of 50 per cent.

Some thermoplastic materials are, however, quite insensitive to changes in humidity. Polystrene and methyl-methacrylate resins are examples of materials having relatively low response to changes in humidity. Thermosetting materials show effects similar to cellulose acetate.

Air Conditioned Laboratory

For the past three years mechanical tests of plastics have been conducted at the University of Illinois as a part of the work of the engineering experiment station in the department of theoretical and applied mechanics. An air conditioned plastics testing room was built in Talbot laboratory. This room has about 200 sq ft of floor space and contains a creep rack for creep tests of 30 specimens; six fatigue testing machines; a combination tension, compression, and torsion testing machine for static tests; an impact testing machine; experimental equipment for repeated impact testing; a lathe; and an analytical balance — a rather crowded laboratory.

Some of the tests are of long duration, particularly the creep tests and fatigue tests. Creep tests have been conducted continuously over a period of 18 months. A fatigue test of a single specimen may last for as long as one month. Thus it is apparent that the air conditioning equipment must be capable of maintaining constant temperature under all atmospheric conditions the year around and, to avoid interruption of the temperature and humidity during tests, the equipment must be fully automatic.

In order to provide constant temperature and constant relative humidity the year around, there are four components which must be controlled—heating, cooling, humidifying, and dehumidifying. In this laboratory, heating and cooling are controlled by a two stage thermostat. Electric strip heaters are used for heat and a "Freon" refrigeration system for cooling. Humidification is accomplished by means of a water spray controlled by a solenoid valve. Dehumidification is handled by condensation on the evaporator of the refrigeration system.

By this method, both humidification and dehumidification require reheating of the air. This is taken care of automatically by the thermostat. Since the refrigeration system is used both for cooling and dehumidifying, rehumidification is occasionally required. This condition is automatically controlled by the humidistat.

In order to avoid excessive bucking of opposing cycles of operation, a thermo-time delay switch is used to introduce a delay of about 5 min between demand and supply on both the humidification and the dehumidification cycles.

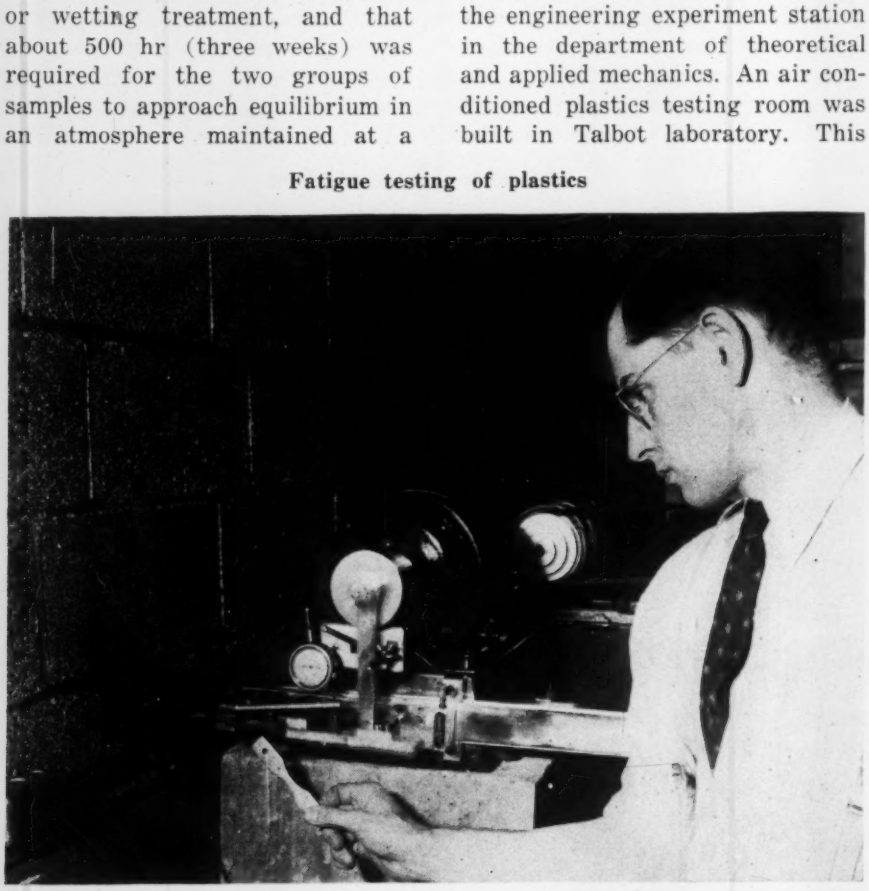
Air distribution, a difficult problem in a crowded laboratory, is handled with reasonably satisfactory results by the use of desk type circulating fans.

Why and How

In the testing of plastics, there are a number of variables which are of extreme importance—the most important are temperature, relative humidity, and testing speed. Thus, the testing engineer studying plastics must employ air conditioning as one of his tools.

Mechanical tests of plastics have been conducted at the University of Illinois for the past three years, as part of the work of the engineering experiment station in the department of theoretical and applied mechanics. Mr. Findley tells here why temperature and humidity control is needed and how it is accomplished.

Fatigue testing of plastics



AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

Journal Section

1895 Fiftieth Annual Meeting 1944

The Fiftieth Annual Meeting of the Society will be held at Hotel Pennsylvania, New York, January 31, February 1 and 2, 1944. This will be an occasion for honoring its founders and those charter members who are still alive.

It was in 1894 that a small group of nationally known engineers, educators and manufacturers met to organize the ASHVE for the purpose of improving the art through the interchange of ideas, and stimulation of scientific research and invention. A preliminary meeting attended by 15 men was held August 2, 1894 at the World Building and on September 10, 75 men met at the Broadway-Central Hotel and elected officers to serve until the First Annual Meeting January 22-24, 1895.

These men realized the basic importance of heating and ventilating as a primary element in the well being of civilized mankind. They knew that the methods and equipment of their day could be improved even beyond their own vision. They foresaw the need of research and one of the first acts of the newly organized Society was to establish a Committee on Standards. It was J. J. Wilson who made the first suggestion that the Society establish a research or testing laboratory. That the charter members had great faith in their enterprise is evident in the discussions at the first Annual Meeting, although little did they dream that progress would be so rapid in their profession.

During the intervening years since that little group of 75 pioneers unfurled the banner of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—3,100 of the real leaders of thought and action in heating, ventilating, and air conditioning have gathered about that standard and carried it proudly before them far along the way of real accomplishment. They may be identified among engineering groups by the distinctive emblem which was adopted by the charter members.

For the 50th Annual Meeting the Meetings Committee is planning a special program that will deal with the past, present and future.

The New York Chapter of the Society as hosts for the occasion feels the full responsibility of its role, and arrangements for the meeting will be handled by a special committee of which Alfred J. Offner, consulting engineer, is General Chairman. Vice-Chairmen are: R. H. Carpenter, J. C. Fitts, W. E. Heibel, C. S. Koehler, Capt. A. E. Stacey, Jr., and R. A. Wasson, President of New York Chapter, ex-officio.

The chairmen of the committees in charge of the special events are as follows: Alfred Engle, Banquet; H. S. Wheller, Hospitality; W. M. Heebner, Finance; A. C. Buensod, Special Events; Mr. and Mrs. H. J. Ryan, Ladies; C. S. Pabst, Inspections: R. V. Sawhill, Publicity; and E. J. Ritchie, Sessions.

A GREAT PAST A GREATER FUTURE

Performance of a Residential Panel Heating System

By H. F. Randolph* and J. B. Wallace,† Utica, N. Y.

SUMMARY — A residential heating system using a warm air ceiling panel is described. Observations are reported for constant and intermittent operation. The calculated panel size based on heat loss of building met the observed requirements. Comfort conditions were maintained, without altering the thermostat setting, through a wide range of fluctuation of outdoor temperatures, due to narrow limits within which mean radiant temperature was maintained. The temperature of floors above the basement increased as the outdoor temperature dropped and was higher than the temperature of the air above the floor.

THE OBJECTS of this investigation were to compare calculated requirements of a panel heating system using warm air as the heating medium and ceiling panels as radiators, with actual operating requirements and to observe the environment produced by such a system under both constant and intermittent operation.

Equipment and Limitation of Use

In March 1941 the erection of a typical two-story and basement eight-room frame house (Figs. 1 and 2) was completed in Utica, N. Y. The heat loss at an 80 F temperature differential was calculated as 44,468 Btu, exclusive of a 12 ft 6 in. x 23 ft 6 in. basement game room and the garage, and 54,277 Btu including the game room. The total floor area of the heated space, including the game room, was 1735 sq ft and the volume 13,052 cu ft and 1442 sq ft and 10,996 cu ft excluding the game room. The exterior wall construction consisted of wood siding on 25/32 in. fiber sheathing board, 2 in. x 4 in. studs, rock lath and plaster. Between the studs, with air space on either side, were placed 2 in. thick bats of balsa wool within vapor seals of asphalt impregnated paper. Between the second floor ceiling joists were placed 4 in. thick bats of rock wool with vapor barrier on the under surface.

All windows and doors were weather stripped. Storm doors and windows were used except on the maid's room and lavatory windows and the front door.

The maid's room was constructed under a sun deck and over a 12 in. excavation below the floor, the area way leading from the excavation to the main part of the basement being closed during all tests. The floor of the maid's room consisted of joists, sub-floor, ½ in. plywood and linoleum with no insulation. The four walls, floor and ceiling of the bath room, and floor and two walls of the kitchen were covered with linoleum.

The house, with no other structure or trees closer than 200 ft and unfurnished and unoccupied, was affected to a greater degree by solar radiation than if protected externally and having the windows fitted with shades and curtains or draperies. During one period of observation, on a clear bright day, the thermostat became satisfied at 6:48 a.m. when the outdoor temperature was 15 F and did not again call for heat until 2:49 p.m. when the outdoor temperature was 30 F. During that period of seven hours, effect of solar radiation through windows of the dining and living rooms satisfied the thermostat while the temperature at the 30 in. level in the bath room, which was unaffected by direct sunshine, dropped from 70 F to 65 F.

Because of such solar effect all tests reported in this paper, unless otherwise noted, were made at night starting at least three hours after sun-down, without lights or other source of heat above the basement than from the heating system.

After the first few observations the game room was omitted from test readings as, while in severe weather its temperature would reach equilibrium with the balance of the house, in mild weather it would be underheated. This apparently was due to the ground temperature adjacent to the three exposed walls of the game room being more constant than the air temperature above ground, resulting in a proportionately higher heat loss from the game room as the weather moderated.

Although temperatures in the locality of this house often drop to —25 F no weather colder than 3 F was experienced during early 1941 or during the 1941-42 heating season when the tests herein reported were made.

Heating System and Method of Control

The heating panels were constructed by stripping, with asbestos tape, the lower edges of ceiling joists and attaching thereto sheets of 30 gage galvanized iron which were also formed down the four walls of each room a distance of 2½ in. Under these sheets and to the joists were attached sheet metal U clips through which were passed 2½ in. sheet metal strips to act as guides for the air. Through perforations in the clips and strips were passed wires by means of which metal lath was suspended. Conventional plaster was applied. Thus a 2½ in. air space above the plaster ceiling was provided.

Introduction of the air to each panel (Fig. 2) was made adjacent to an exposed wall so that the warmest part of each panel was directly over the area of greatest heat loss.

Heat was supplied by a gas fired hot water boiler connected on the supply side through a flow control valve to a copper heat exchanger and back to the boiler through a circulator (Fig. 3). The exchanger and

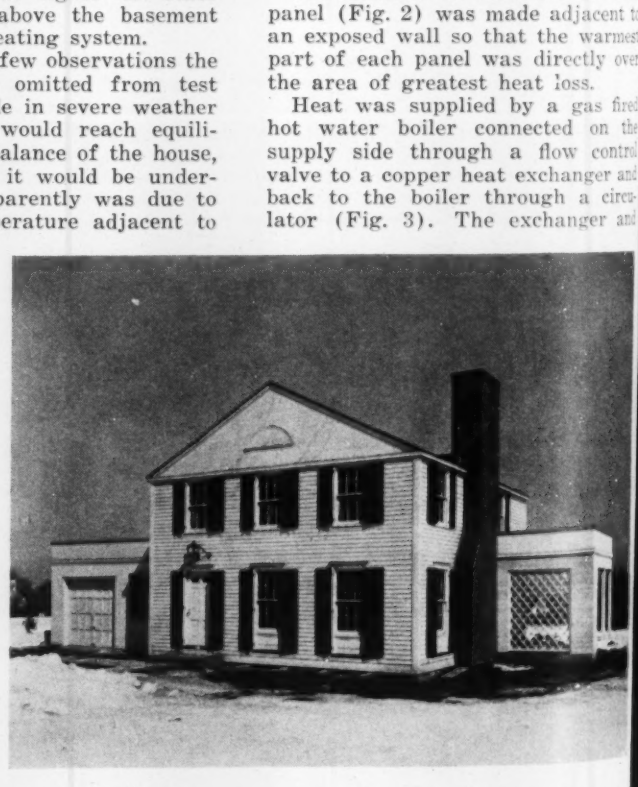


Fig. 1—House in Utica, N. Y., where ceiling panel heating studies were conducted

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†Research Engineer, International Heater Co.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Pittsburgh, Pa., June, 1943.

a 14 in. squirrel cage blower were installed in a galvanized iron duct system with one supply and one return riser to each ceiling panel so that the air was circulated in a closed system over the exchanger through the ducts to the panels and returned, with none of the air being projected into the living quarters of the house. The boiler-exchanger combination was used, rather than a direct fired unit, merely to provide a flexible means of controlling air temperature for test purposes.

From the basement trunk lines 10x3¼, 12x3¼ and 14x3¼ riser stacks were run to and from the various panels. One supply riser and one return riser were run to each room except the bath and second floor hall in which two rooms were heated from one set of risers. Balancing dampers were installed in the basement piping, and control dampers in the return riser from each bedroom to provide means within the room for shutting off the source of heat.

A conventional heat anticipating thermostat was installed at the 5 ft level on the west wall of the dining room, wired through a relay to the motors driving the blower and water circulator. The gas burner was controlled by an aquastat immersed in the water of the boiler.

Upon a call for heat by the thermostat the blower and circulator started simultaneously, delivering the warmed air to the panels, and on a 10 F drop in water temperature within the boiler the gas burner started, continuing to run until the 10 F had been restored. Upon satisfaction of the thermostat the blower and circulator stopped.

Instrumentation

During construction of the house 176 24-gage copper constantan thermocouples were installed as follows:

21 in seven groups of three each for exposed walls gradients at 6 ft and 16 ft levels above grade on the south, east, and north walls and at the 16 ft level of the west walls one of each group embedded in the plaster; one on the outer face of the insulation and one embedded in the siding.

Fig. 2—Floor plan showing heating panel arrangement

33 in 11 groups of three each for ambient readings located at different levels for different tests. One 3 in. or 6 in. above the floor, one at the 30 in. or 60 in. level and one 3 in. or 6 in. below the ceiling in each room except the lavatory. No thermocouples were shielded.

50 in 10 groups of five each embedded in the ceiling plaster for heating panel readings. One 15 in. from each corner of each panel and one in the center of each panel.

45 in nine groups of five each embedded in the finish floor for floor readings. One 15 in. from each corner of the room and one in the center of the room. No floor thermocouples were installed in the northeast bedroom.

24 in 12 groups, one at the entrance to and one at the discharge from each panel for readings of the inlet and discharge air temperatures.

3 embedded in the plaster of the north wall of living room at the 5 ft level, one opposite a supply riser, one opposite a return riser and one with no riser behind.

All thermocouples were connected to a central switchboard and potentiometer located in the basement (Fig. 4). A four point recording thermometer was used for continuous records of ambient readings in the three bedrooms and bath on the second floor.

Test Observations

Control System

When the house was erected it was anticipated that, with intermittent operation of the heating system, the thermal storage capacity of the wire lath and ¾ in. plaster comprising the ceiling panels might be of a magnitude that would result in objectionable temperature fluctuations within the living quarters. To determine this, observations were made over a period of 12 hours with the thermostat set at 70 F and with readings taken at the 5 ft level every 30 min, and 6 in. above the floor and 6 in. below the ceiling every two hours in the ten rooms of the first and second floors. The outdoor temperature averaged 12 F.

During the 12-hour period there were 26 cycles of operation totaling 3 hours 49 min. The average of the *on* periods was 8.8 min and of the *off* periods, 17.6 min.

The average of all readings 6 in. above the floor (Fig. 5) was 67.5 F and the maximum variation below this was 0.6 F and above, 0.4 F. The average of all readings at the 5 ft level was 70.7 F and the maximum variation below this was 0.7 F and above, 0.4 F. The average of all readings 6 in. below the ceiling was 72.0 F and the maximum variation below this was 0.4 F and above, 0.3 F.

As these observations did not indicate any objectionable temperature fluctuations, and because of the physical impossibility of recording readings frequently enough at all stations to obtain the temperatures at the exact times the thermostat called for heat and became satisfied, a test was made at a 30 F outdoor temperature of one room only. Readings were taken in the living room 3 in. and 30 in. above the floor and 3 in. below the ceiling at the moment the ther-

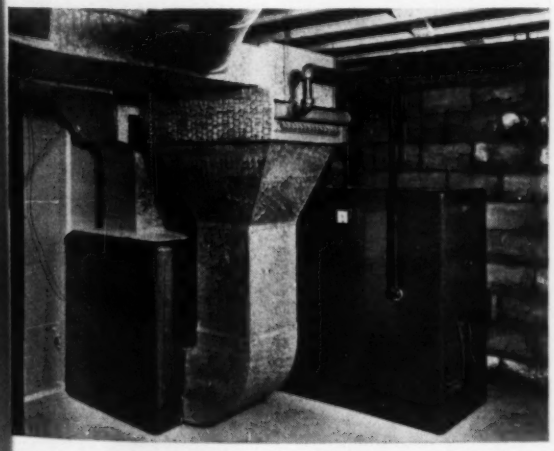

Fig. 3—View of boiler, blower, and heat exchanger

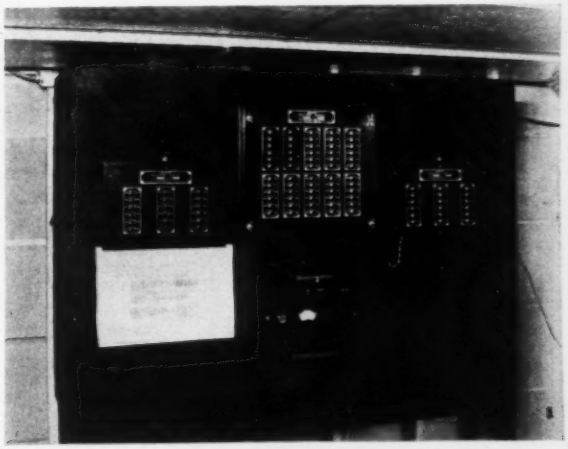

Fig. 4—View of switchboard and potentiometer

mostat called for heat and blower started and at the moment of thermostat satisfaction when the blower stopped. The test was conducted through a period of five cycles of system operation from 10 p.m. to 2 a.m. The average of the *on* periods was 7.3 min and the average of the *off* periods 33 min.

During the test period the greatest temperature variation at any one point was 1 F and the average temperatures at the beginning of the *on* period were 0.2 F higher than at the end of the *on* period. This result, being the reverse of expectancy, indicated the thermostat was affected by radiation and stopped the source of heat before the thermocouples registered a temperature rise and that subsequent heat dissipation from the panel resulted in an increase in air temperature during the *off* period.

To compare this intermittent operation with the possible over-heating effect of a prolonged period of operation the house was cooled, on a day when the outdoor temperature was 16 F, to an average ceiling temperature of 60 F, floor of 59 F, inside wall 56 F, exposed wall 54 F and 30 in. level of 55 F.

At 12:20 a.m. the heating system was started and continued in operation for 4 hours and 25 min before the thermostat became satisfied. During that time the average temperature of the air leaving the exchanger in the basement was 130 F, entering the panels 123 F and leaving the panels 85.4 F, these latter two figures being the average of 120 readings. The air temperature drop per lineal foot of duct averaged 0.22 F, and the air temperature drop through the panels averaged 37.6 F.

For a period of 50 min after thermostat satisfaction the temperature

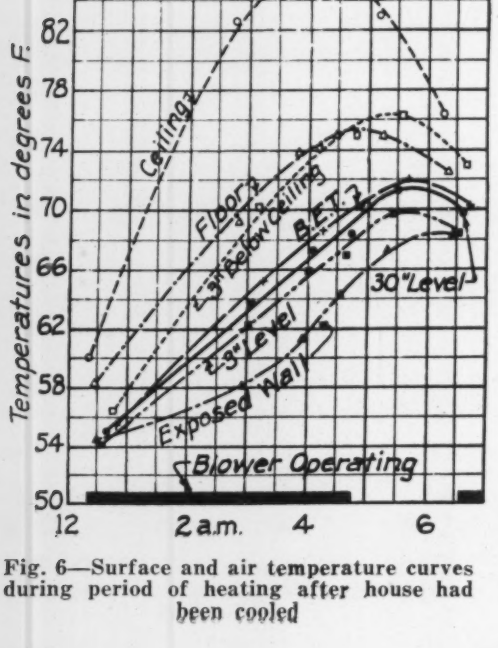

Fig. 6—Surface and air temperature curves during period of heating after house had been cooled

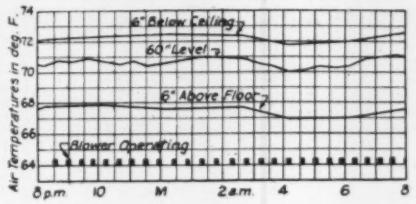

Fig. 5—Graphic log of average temperatures at three levels with intermittent operation. Outdoor temperature, 12 F

of the air and all surfaces of the room continued to rise with the exception of the ceiling which cooled rapidly. In Fig. 6 these temperatures and the calculated British equivalent temperature are plotted against time. During the 50-min temperature rise subsequent to the heat source being stopped the calculated British equivalent temperature rose only 1.8 F and during the succeeding 60 min fell to a value closely approximating that at which the thermostat became satisfied and at which heat was again required. Air temperatures at the 3 in. and 30 in. levels were higher at 6:35, at the beginning of an *on* period than at 4:45, the beginning of the *off* period.

The temperature variation during all of these periods of observation was such that it was not deemed necessary to use a more elaborate control system so as to provide constant operation with the temperature of the heating medium regulated inversely with the outdoor temperature.

Panel Temperatures

Examples of calculating radiant heating are given in ASHVE Guide[1] which indicate agreement between two different methods. In one example the panel area and temperature required to maintain a 71 F mean radiant temperature is computed from the difference between the total radiation of all interior surfaces at 71 F, and the total radiation of such surfaces at their calculated or assumed temperatures, with an indoor temperature of 65 F and 0 F outdoors. In another example the heat loss of the same room is computed in the conventional manner according to The Guide 1943, Chapter 6, Heating Load. The British thermal unit requirements in both cases are approximately the same.

If the room used in these examples were partitioned into two rooms the British thermal unit requirements in the first example would be increased by the amount necessary to compensate for the difference in radiation from both sides of this partition at its desired temperature of 71 F and assumed temperature of 60 F. There would be no increase in the British thermal unit requirements computed by the second method as the partition would create no additional heat loss.

Applying the first method of calculation to this particular house the total heat emission from all interior surfaces, exclusive of the game room, as shown in Table 1, was 777,717 Btu per hour, equivalent to a mean radiant temperature of 61.8 F with emissivity of 0.94.

The inside surface temperatures were developed as follows: Outside wall, glass and exposed floor from Fig. 7, p. 810 of The Guide 1943. Inside wall was assumed to be no lower than the exposed wall, although footnote with Table 3, p. 812 of The Guide 1943 assumes a temperature for such a surface as 60 F which

[1] Heating Ventilating Air Conditioning Guide 1943, Chapter 45, Radiant Heating.

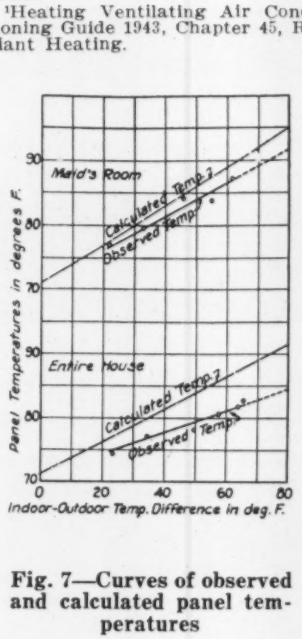

Fig. 7—Curves of observed and calculated panel temperatures

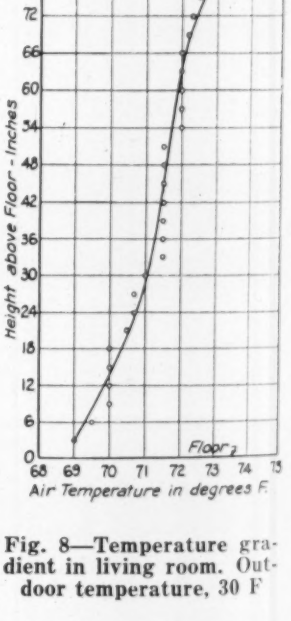

Fig. 8—Temperature gradient in living room. Outdoor temperature, 30 F

Table 1—House data for calculating panel temperature based on 65 F inside air; 71 MRT; 15 mph wind; —10 F outside temperature

Surface	Area Sq Ft	U	Estimated Inside Surface Temperature Deg F	Emissivity e	Heat Emission Btu per Sq Ft per Hr	Total Heat Emission Btu per Hr
Outside Wall	1254	0.083	62	0.95	121.4	152,236
Glass, Single	48	1.13	16	0.90	79.6	3,821
Glass, Double	262	0.75	32	0.90	90.9	23,816
Inside Wall	2058	—	62	0.95	121.4	249,841
Unexposed Ceiling	641	—	67	0.95	125.9	80,702
Exposed Ceiling	848	0.065	67	0.95	125.9	106,763
Exposed Floor	118	0.294	52	0.93	110.1	12,992
Unexposed Floor	1260	—	60	0.93	117.1	147,546
Total	6489			Avg. 0.94		777,717

assumed temperature was used for the unexposed floor. The estimated temperature of the exposed ceiling from Fig. 7, referred to, would be approximately 63 F x 1.06 = 67 F and the temperature of the unexposed ceiling was assumed at the same value.

For a mean radiant temperature of 71 F, having a heat emission of 128.3, the total emission for all surfaces would be 6489 x 128.3 = 832,539 Btu per hour or an additional requirement of (832,539 — 777,717) 54,822 Btu per hour. Dividing 54,822 by the panel area of 1267 gives 43.3 Btu per square foot per hour which, added to 125.9 Btu per hour, the emission of the unheated ceiling, equals 169.2 Btu per hour requiring a panel temperature of approximately 107.1 F.

Computing the heat loss of the house, exclusive of the game room, according to Chapter 6, of The Guide 1943, including infiltration but excluding heat loss upwards from panels, the British thermal unit loss per hour at an 80 F temperature difference was, as shown in Table 2, 39,984. Dividing this by the panel area of 1267 gives 31.6 Btu per square foot per hour. As approximately 70 per cent of the heat from a horizontal panel with heat flow downward is by radiation, the Btu per square foot would be 31.6 x 0.70 = 22.1 which, added to 129.6, the radiation from a surface at 71 F (the desired mean radiant temperature) with an emissivity of 0.95, gives a total radiation of 151.7 Btu equal to a panel temperature of 91.6 F. This temperature agrees with the average for the house when each room is calculated individually and it is this temperature that is used in the following comparison with test results.

To compare the calculated panel temperatures with actual requirements tests were run at various outdoor temperatures. For these tests the water temperature in the boiler was reduced to a point that required constant operation of the heating system. Readings were taken at all stations and each test was run over a sufficient period of time (generally 5 to 6 hours) to insure temperatures throughout the house being stabilized as indicated by two successive readings at 60-min intervals being in close agreement. In Fig. 7 the panel temperatures are compared with the calculated requirements. Each point plotted represents the sum of the products of the average of 5 points in each room panel multiplied by its area, divided by the total panel area.

Extrapolating the observed temperature curve to the 80 F temperature

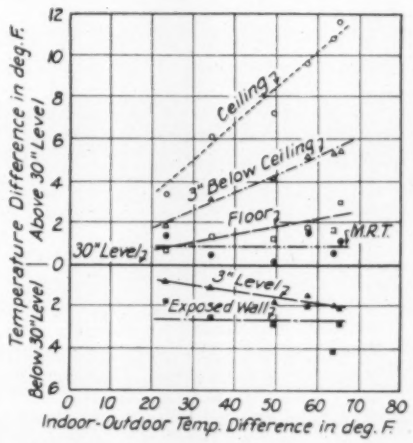

Fig. 9—Surface, air, and MRT curves

difference gives a required panel temperature of 85 F which is 6.6 F below the calculated temperature of 91.6 F or 68 per cent of the calculated rise above 71 F based on the heat loss method of computation and 38.8 per cent of the calculated rise above 71 F based on the radiation method of computation. This discrepancy between the observed panel temperature and that calculated by the heat loss method might be accounted for by stray heat loss from the heating system and chimney. At design temperatures with an 0.25 F drop per lineal foot of ducts to panels, the heat dissipation from the supply side of the heating system alone would approximate 16 per cent of the calculated requirements. If the heater, return system and chimney accounted for a similar amount, and if the required heat input from the panels were computed less this total allowance, then the observed and calculated panel temperatures would be in close agreement.

The calculated average panel temperature for the house was 91.6 F but for the maid's room only it was 95 F. Into this room there was less stray heat than into any other room, as it was unaffected by ducts or risers other than to itself. Fig. 7 also compares the panel calculated temperatures with the test results in this room only where the recorded panel temperature rise above 71 F was 88 per cent of the calculated requirement.

Temperature Gradient in Living Room

Observations of the gradient in the living room were made on two occasions, one week apart, with outdoor temperatures of 30 F and 30.5 F. One of the thermocouples attached to the standard reaching from floor to ceiling in the center of the room was moved in 3 in. increments in both directions between floor and ceiling and the temperature at each station recorded. In that way four sets of readings were taken over a total expired time of 6 hours. The average of all these readings, plotted on Fig. 8, is 71.48 F. The average of stations 3 in. and 30 in. above the floor and 3 in. below the ceiling is 71.5 F.

The temperature variation in the living room between 3 in. above the floor and 3 in. below the ceiling was of the order of 5.5 F.

Air Temperatures Compared with Mean Radiant Temperatures

As no means of measuring mean radiant temperature was used, it could only be arrived at from readings of 105 thermocouple locations in floors, walls and ceilings, and calculating the glass temperatures from heat transmission coefficients, and assuming partition surface temperatures to be the average of air temperature readings. The average temperature of these inner partitions, with the system of heating employed, probably exceeded the average air temperature due to heat transmission

Table 2—Heat loss calculation, not including heat loss upward from panels

Surface	Area Sq Ft	U	Temperature Difference	Btu
Outside Wall	1254	0.083	80	8327
Glass, Single	48	1.13	80	4339
Glass with Storm Sash	262	0.75	80	15720
Exposed Ceiling, No Panel over	111	0.065	80	577
Exposed Floor	118	0.294	40	1387
				30350
Infiltration	283.5 ft[a]	23.6 cfm	0.018 × 80	9634
				39984 Total

[a] 100 per cent of crack of windows, 200 per cent of crack of doors on north and east exposures.

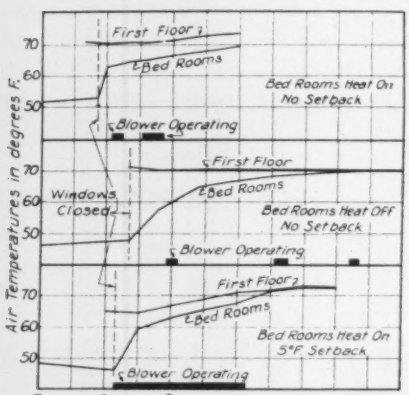

Fig. 10—Graphic log of temperatures at 30 in. level during morning pick-up

from riser pipes within the partition. Three thermocouples located on the north wall of the living room indicated an average surface temperature of 1.8 F above air temperature with no riser behind, 4.3 F higher with a return riser behind and 7 F higher with a supply riser behind. If this is indicative of all the partitions, the mean radiant temperature reported herein is slightly lower than that which actually existed.

F. C. Houghten and others have shown[2] that in a room heated by free standing radiation the mean radiant temperature never exceeded the air temperature and readings plotted in Fig. 9 indicate a similar condition with this particular application of ceiling panel warming.

The curves in Fig. 9 showing temperatures of the floor, 3 in. above the floor, 30 in. above the floor, 3 in. below the ceiling, the ceiling, exposed wall and calculated mean radiant temperature are plotted from tests run at outdoor temperatures ranging from 4 F to 47 F with the heating system under constant operation for a sufficient period of time to reach equilibrium as indicated by two successive readings at 60-min intervals. The calculated average mean radiant temperature for all tests exceeded the temperature at the 30 in. level by 0.9 F and likewise the average of the air temperatures 3 in. and 30 in. above the floor and 3 in. below the ceiling was 0.9 F above the 30 in. level. If the mean radiant temperature were compared only with the air temperature below the breathing line then the mean radiant temperature would be of a slightly higher value.

This system employed horizontal panels with heat flow downward from which heat transmission by convection is less than from vertical panels or from horizontal panels with heat flow upward. It is doubtful, therefore, in any well constructed panel heated house having no more than a normal number of air changes that the air temperatures would ever be depressed appreciably below the mean radiant temperature.

With radiator and convection residential heating systems the gradient between the 3 in. and 30 in. levels has been found to increase with an increase in the indoor-outdoor temperature differential.[3] While a similar condition existed in this investigation the floor temperatures, as shown in Fig. 9, not only exceeded the air at the 30 in. level but also the floor temperature increased with the indoor-outdoor differential.

As a drop in outdoor temperature was accompanied by an increase in ceiling and floor temperatures, a rather constant exposed wall temperature and a decreased glass temperature, the mean radiant temperature held within such narrow limits that comfort conditions were maintained during all outdoor conditions without a change in thermostat setting.

Morning Pick-Up

Three tests were run to observe conditions existing in the morning after bedroom windows, which had been opened during the night, were closed. In the first test the heat remained on in the bedrooms, the thermostat was not set back, the bedroom doors were closed and windows were opened 3 in. from the top and 6 in. from the bottom. The second test was made under similar conditions except that heat in the bedrooms was turned off. In the third test heat

[2]ASHVE Research Report No. 1172—Radiation as a Factor in the Sensation of Warmth, by F. C. Houghten, S. B. Gunst and S. Suciu, Jr. (ASHVE Transactions, Vol. 47, 1941, p. 117.)

[3]Operation of the Research Home with Reduced Room Temperatures at Night, by A. P. Kratz, W. S. Harris and M. K. Fahnestock. Digest of Research, Engrg. Experiment Station, University of Illinois. (Journal Section, Heating, Piping & Air Conditioning, December, 1942.)

Fig. 11—Reproduction of four-point recorder chart comparing temperature at 30 in. level in bath with three bedrooms in which windows were open during night and heat off

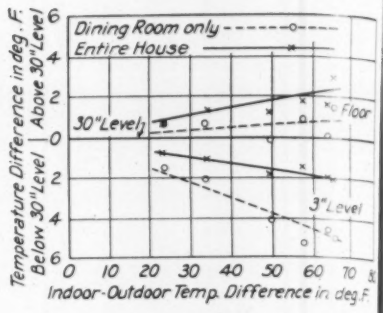
Dining Room only
Entire House
30" Level
Floor
3" Level
Temperature Difference in deg. F. Below 30" Level | Above 30" Level
Indoor-Outdoor Temp. Difference in deg. F.

Fig. 12—Curves of floor and air temperature at 3 in. level in dining room and entire house

remained on in the bedrooms but the thermostat was reduced 5 F during the night. In all three tests the outdoor temperature ranged from 30 F to 34 F. These three tests are plotted in Fig. 10.

In the first test with heat remaining on in the bedrooms and no night set-back, the bedroom temperatures dropped to an average low of 51 F. At 7:52 a.m. the bedroom doors were opened and windows closed. On demand from the thermostat the heating system operated from 8:04 to 8:15 and from 8:32 to 8:52 at which time the second floor bedrooms were 5 F lower than the first floor. The heating system did not operate again for several hours due to sun effect.

In the second test, in which heat in bedrooms was turned off but with no set-back, the bedrooms reached a low average temperature of 45 F. At 8:30 a.m. the bedroom doors were opened, windows closed and dampers opened in the risers. On demand from the thermostat the heating system operated from 8:55 to 9:05 — from 10:33 to 10:45 and from 11:41 to 11:50 or a total of 31 min out of 2 hours and 30 min. At 11:50 a.m. the temperatures of bedrooms had equalized with the first floor.

Fig. 11 is a reproduction of the chart from the 4 point recorder from 7:00 a.m. to 1:00 p.m. Temperature of the bathroom held within 1 F of 70 F until 8:30 a.m. when the bedroom doors were opened. The bathroom temperature then dropped for about one hour while the bedroom temperatures rose. All of them equalized at 11:50 and reached and continued at 70 F at 12:30 p.m.

In the third test heat remained on in the bedrooms but the thermostat was reduced 5 F. During the night the mean bedroom temperature dropped to 46 F. The doors were opened, windows closed and thermostat advanced 5 F at 8:08 a.m. at which time the heating system started and continued in operation until 10:08 a.m. when the temperatures of both floors were practically equalized. For an hour thereafter temperatures continued to rise — due partially, at least, to solar radiation. Under normal operation, however, the thermostat would have been automatically

advanced at least one hour before the bedroom doors were opened so that any overheating would be minimized.

Effect of Large Glass Areas

Exposure of the dining room consisted of 115 sq ft of wall and 77.3 sq ft of glass and doors, the windows and french doors taking up 12 lineal feet of wall at the floor line.

A comparison of floor temperatures and air temperatures 3 in. above the floor in the dining room and the average of the house is shown in Fig. 12. These curves were plotted from readings of constant operation tests at various outdoor temperatures.

The variation between the 3 in. and 30 in. levels in the dining room was 137 per cent greater than in the house as a whole due apparently to the convection currents established by the large glass area. Because the floor was warmer than the air at either the 3 in. or 30 in. level, the lower air temperature was not of a magnitude to seriously affect comfort but it does suggest the same consideration for glass placement with panel heating as with any other.

Summary

1. A basement game room from which the heat loss fluctuation was not uniform with the balance of the house, could not be satisfactorily heated in mild weather.

2. The combined thermal capacity of air as the heating medium and normal plaster ceilings as the heating panels resulted in such rapid response to heat demand that comfort conditions were maintained with intermittent operation of the heating system under control of a conventional heat anticipating thermostat.

3. In this house with normal air changes, and warmed by a panel heating system, the air temperature tended to equalize with the mean radiant temperature.

4. Calculated panel temperatures based on the heat loss of the structure more closely approached observed requirements than the calculated temperatures based on radiation.

5. Stray heat loss into the house from chimney, heater and duct work provided a substantial factor of safety.

6. The mean radiant temperature was held within such narrow limits over a wide range of outdoor temperatures that comfort conditions were maintained without altering the indoor air temperature.

7. After a reduced night temperature the extended period of heater operation, required to restore day time temperature, reduced the time required for temperature equalization of rooms, in which windows were opened during the night, with the remainder of the house.

8. Floors above the basement increased in temperature as the outdoor temperature dropped and temperature of such floors exceeded the temperature of air adjacent to them. This higher floor temperature tended to counteract the affect of convection currents and depressed low level air temperatures established by large exposed glass areas.

Discussion

T. Napier Adlam, New York, N. Y. (Written): It appears from the authors' statement that the instrument used to record the air temperatures did not indicate any great change over a given period—indicating that the instruments used were not capable of registering comfort conditions.

The air temperatures recorded being very much higher than necessary with a true radiant heating system would disturb the balance of heat loss from the human body. While the calculations called for a panel surface temperature 107.1 F. the average surface temperatures of the panels were probably 91.6 or 95 F. This can be accounted for by the extra heat coming from the floors and warmed portion of the walls, also by the fact that while the calculation made for the building was based on an outside temperature of —10 F, the test was made with an outside temperature of +12 F.

F. F. Giesecke, Urbana, Ill. (Written): The authors have made a valuable contribution to the panel heating literature. They have developed a panel heating system which is new and simple, and have shown that, with this system the temperature can be controlled satisfactorily with the conventional thermostat. They have also supplied experimental data which can be used for further studies of panel heating. For example, Fig. 9 shows that the MRT was about 1 deg above the 30 deg level air temperature.

This is evidence that, in any one room, the air temperature and the MRT are interdependent. If a room is heated by means of panels until its thermal conditions are constant, the MRT will be slightly higher than the mean air temperature, because the air is heated by its surroundings.

If a room is heated by means of a convector until its thermal conditions are constant, the MRT will be slightly lower than the mean air temperature, because the heat transfer is from the air to its surroundings.

In the case of the building studied by the authors, it is impossible to design the heating system so that there will be a definite MRT and also a definite mean air temperature unless the two selected temperatures happen to correspond to each other. For example, it is evident from the authors' experimental data that when the MRT was 71 F, the mean air temperature was about 70 F and could not have been reduced to 65 F, unless cool air was continuously blown into the room in sufficient quantity to lower the air temperature from 70 to 65 F.

In the authors' system, the ceiling panel was heated by blowing the air along the *upper* surface of the panel. It would be of interest and of value if the authors could continue their studies by heating the ceiling panel with air blown *across the room* along the *lower* surface of the panel. To secure design data for such an installation, tests were conducted in a radiator test booth at the University of Illinois. While a radiator was being tested, the ceiling temperature was about 93 F and the air temperature, 3 in. below the ceiling, was about 100 F, or 7 deg higher. After the radiator test had been completed, the test booth was closed and allowed to heat until the ceiling temperature was about 101 F; at the same time the air temperature was about 111 F, or 10 deg higher. The heat was then cut off and the booth allowed to cool until the ceiling temperature was about 95 F at which time the air temperature was about 100 F, or about 5 deg higher.

It appears from this test that a ceiling temperature of 86 F in the booth could be easily attained with an air temperature of 93 F, 3 in. below the ceiling.

In the authors' tests, it appears from Fig. 6 that a ceiling temperature of 86 F was secured by blowing air over the ceiling at a mean temperature of about $\left(\frac{125 + 85}{2}\right) =$ 105 F, or about 12 deg higher than necessary if the air were floated along the underside of the ceiling panel. A system constructed and operated as suggested would be lower in cost and possibly also lower in the cost of operation than the system used by the authors and, in addition, it would permit partial purification and partial humidification of the air, so that it might be classified as a combination panel heating and air conditioning system. I hope the authors will consider these suggestions sympathetically and present another paper in the future.

A. J. Johnson, Philadelphia, Pa. (Written): The research of the authors has undoubtedly added invaluably to our fund of domestic heating knowledge. The paper is so clearly presented and the subject so new that there seems to be little room for extended discussion at this time.

There is one point, however, which might warrant further consideration. The paper mentions the fact that it required 4 hours and 25 min to raise the temperature from 60 deg to a point where the thermostat was satisfied. It later refers to 2 hours of continuous heater operation for recovering 5 deg in the morning. Both of these acceleration periods would seem to be rather long, which suggests that with a low temperature, concentrated heating medium, increased attention might well be given to air circulation and thermostat placement.

A point that is not always realized is that an eight-room house contains over a ton of air, which must be placed in continuous motion in order to secure uniform heating. Since the only motive power after the fan has stopped is the source of heat supply, it follows that when this is either l w in temperature or concentrated in area more time is required to overcome the inertia of the air than otherwise. This would account for delayed acceleration, over-riding in rooms adjacent to the thermostat as mentioned in this paper, and a tendency toward difficulty in balancing the temperatures in the several rooms.

Several suggestions for increasing the comfort in a house of this type are therefore indicated:

1. It might be advisable to study the relative fuel consumption with uniform 24-hour temperatures as compared with depressed night temperatures. If it can be shown, as might be suspected, that in any house with a system requiring prolonged acceleration periods, 24-hour comfort does not involve appreciable additional fuel, the increased comfort factor is obvious.
2. In designing a low temperature source of heating, it would seem to be in order to carefully study the path of convected currents through the house, so that each heating panel will in effect relay the normal flow of heated air to the next panel rather than setting up localized circulation or opposing currents.
3. A change in the location of the room thermostat might be in order. With intermittent types of heat there is a tendency for the room in which the thermostat is located to become heated before the circulation of air throughout the house is normal. Thus, instead of the design condition of each radiator or panel satisfying its own area there is necessarily a secondary effect, due to the cooler rooms having to draw upon the heated air in the warmer rooms before stabilization has been achieved. This condition can probably be overcome by a very painstaking balancing of the entire heating system. However, a simpler solution might be found if the thermostat could be located in point of average temperature such as for example an upstairs hall.

F. C. Houghten, Pittsburgh, Pa. stated that from a theoretical as well as practical viewpoint the addition of a partition to a given space would increase the heat loss because the partition would take on a higher temperature than the exposed surface, would have a higher mean radiant temperature, and would therefore, radiate more heat to the inside surfaces of the exposed wall. He considered the ASHVE Guide method of computing the radiant panel size to be theoretically correct.

C. Tasker, Toronto, Ont., Canada, suggested that there was need to standardize the method of measuring radiant heat in order that field test results of different investigators might be comparable.

P. F. Gordon, Bloomfield, N. J., asked, (1) how much insulation was used between the first floor ceiling and second floor construction, and (2) whether the heat flow through the second floor construction entered into the discrepancy between the observed and calculated mean radiant temperature. He thought that the observed ceiling panel temperature might be lower than the calculated value because the first or second floor construction temperature and the plaster surface temperature of the interior walls might have been higher than calculated due to conveyance of air up through ducts in the walls. It was probable that the air temperature could not be held below the mean radiant temperature because a lot of surface at points other than the ceiling was effective as radiant heating surface.

L. T. Avery, Shaker Heights, Ohio, inquired whether the relative humidity had been determined when the conclusions were reached regarding comfort.

John James, Cleveland, Ohio, asked how the authors had secured uniform distribution of air flow in the thin ceiling panels and whether special precautions were taken to prevent discoloration or cracking of the plaster.

Elliott Harrington, Schenectady, N. Y., stated that he would be interested in the authors' impression of the comfort characteristics or operating advantages of panel heating.

Ferdinand Jehle, Indianapolis, Ind., asked whether warm air panel installations would usually employ a forced warm air furnace and if so whether the air temperature in the panel could be controlled with sufficient accuracy.

C.-E. A. Winslow, New Haven, Conn., pointed out that the paper had shown two advantages that might be expected from radiant heating by a ceiling panel: (1) the differential between floor and ceiling was small as compared with most convection systems; and (2) there was less air movement.

Convection heat loss from the body depends on the mean surface temperature of the body and the mean air temperature multiplied by the square root of the air movement. At a mean velocity of 16 ft instead of 64, for instance, the difference between mean skin temperature and mean air temperature can be twice as great with the same sensation of comfort. The evenness of temperature from floor to ceiling and the low air movement permits use of low air temperature with the same comfort.

Dr. Winslow mentioned the successful use of banks of light bulbs against the ceiling and floor for the purpose of rapidly raising room temperature in the morning in the J. B. Pierce Laboratory. Room temperature could be raised from 55 to 70 F in 30 to 45 min by temporarily increasing the wattage input.

Mr. Randolph, referring to remarks by Messrs. Adlam and Tasker, stated that no standard of measurement for registering comfort exists at the present time. He suggested, in reply to Mr. Adlam's comment, "that with the increasing interest in radiant heating it would be well to have a definition of the term *true radiant* heating system." It is not clear at present whether this term refers to the means by which the heat is introduced into a room, or whether it refers to a relation between air temperature and the MRT.

In reply to Dr. Giesecke's suggestion of using warm air under the panel rather than over it, the author stated, it was interesting and he could see no reason why it would not work. However, he felt, as the air would be at a higher temperature than the panel objectionable dust and dirt streaks would undoubtedly be formed on the ceiling panel.

In answering Mr. Johnson, who requested the time required to recover 5 F in the morning after a night setback, the author stated that the rate of recovery could be shortened by using a slightly higher temperature in the heating medium, but the elapse of time shown in the paper is no greater than that experienced in warm air and radiator heated homes at the University of Illinois. This recovering, the author pointed out, refers only to the air temperature and occupants of panel heated homes apparently experience comfort conditions soon after the heating system is in operation and before any appreciable air temperature elevation has been noticed.

In commenting on Commander Houghten's remarks, the author stated that he agreed with the fact that by running an internal partition in a room the partition would take on a higher temperature than the exposed surface, thereby increasing the MRT, but he did not understand why a greater heat input would be required which is the case when calculated according to Chapter 45 in the ASHVE Guide. He further stated if, in a room 20 ft x 24 ft with the north and south walls exposed and having areas equal to those given in The Guide example, a partition is run from north to south the required heat input would be increased some 15 per cent. It is because of such a condition, he stated, that he believed the heat loss method of calculation should be used.

The author stated that Mr. Gordon is correct in his assumption that floor temperatures were increased as a result of panels or duct work underneath, and likewise as pointed out in the paper partition temperatures were elevated by risers therein. It did not seem to the authors as though the discrepancy between calculated and observed panel temperatures was great when the heat dissipation of the supply side of the system alone accounted for 16 per cent of the heat loss, which was not considered in calculating the panel temperatures. It may be that if heat input had been from the ceilings exclusively the air temperature in relation to the MRT would have been somewhat lower; however, it is the authors' opinion that in an ordinary residence, without ventilation and using low temperature panels, the air will eventually approach the MRT.

The author also pointed out that 6 or 7 installations of the type described in the paper had been made and that all were equipped with direct fired units either gas, oil or stoker-fired and that all had been very satisfactory to the owners. He did not intend to conclude that the warm air panel system was better than a well installed radiator or convection warm air heating system in a well-built house. The ability to place draperies or furniture where desired without need for considering the heating system was an admitted advantage, the author stated.

No cracking of plaster had been observed in the installations made. Three made use of expanded lath with conventional plaster (without binding) and two or three made use of gypsum lath and plaster attached to it. No discoloration had been observed although the panels had been painted to match the rooms. In the installation described the plaster was smooth. Humidity was not determined although its bearing on comfort was acknowledged. He brought out that radiant temperatures were not obtained because the method of taking them had apparently not been standardized.

Mr. Randolph then compared the time of morning temperature pick-up with that reported in tests of a radiator system in the Research Residence at Urbana and pointed out that in the latter two hours were required to recover 6 F and also that the cold walls at the end of this period still had an uncomfortable effect on the occupants.

Use of the Down-Draft Coking Method for Smokeless Combustion

By Julian R. Fellows* and John C. Miles,** Urbana, Ill.

SUMMARY—Application of the down-draft coking principle in design of a smokeless furnace and of a smokeless conversion burner for soft coal is described. Operation results, test data, and smoke observations show successful performance.

THE WELL known fact that the temperature required to ignite a coal gas-air mixture is much higher than that required to convert the volatile matter in bituminous coal to the gaseous state, makes it necessary to use either the down-draft principle or the underfeed principle in the construction of small furnaces designed to achieve smokeless combustion. The underfeed principle has been used to some extent in hand fired furnaces but its use has been generally abandoned because the operation of a furnace equipped with the necessary mechanism for forcing the coal through the bottom of the fuel bed was found to be too difficult for the average housewife to operate. The down-draft principle has also been tried in innumerable forms by many inventors and furnace designers, but for various reasons, such as grate failure, baffle wall failure, arching of the coke, difficulty in removing clinkers and "puff back," practically every attempt to use this principle has resulted in ultimate failure.

A project of the Mechanical Engineering Experiment Station at the University of Illinois started in 1939 by the authors, with the cooperation of Prof. A. P. Kratz, has had for its objective the practicable application of the down-draft principle to hand fired furnaces designed to burn bituminous coal smokelessly. In developing the down-draft coking principle an attempt has been made to eliminate all of the faults that have characterized previous applications of the down-draft principle.

The Down-Draft Coking Principle

Fig. 1 shows a longitudinal vertical section of the body of a furnace incorporating the down-draft coking principle. This figure illustrates the method of burning the fuel, without being concerned either with the arrangement of secondary heating surface or with the use that is made of the heat liberated during the combustion process. The typical furnace consists of a coking chamber, a coke burning chamber, a baffle wall with vertical air passages, a sloping deadplate under the coking chamber, an inclined "pin hole" grate under the forward portion of the coke burning chamber and a horizontal shaking grate under the rearward portion. The baffle wall containing vertical air passages separates the upper portion of the coking chamber from the combustion chamber and directs the gases released from the coking coal over the surface of the incandescent coke in the coke burning chamber. Three definitely proportioned openings admit air in the correct proportions to the secondary air passages, the coking chamber, and the ash compartment below the grates. The secondary air which enters the furnace through ports above the firing door passes through a horizontal passage in the roof of the coking chamber, then through the vertical passages in the baffle wall, emerging from the bottom of the baffle wall, where it is thoroughly mixed with the gases passing toward the combustion chamber from the coking chamber. The primary coking air, in an amount that is just sufficient to support a low combustion rate, enters the coking chamber through a small port in the firing door. The heat liberated by the slow combustion in the coking chamber gradually drives the volatile matter from the fresh coal, thus converting the coal to coke. The primary under grate air which enters the furnace through a small port in the ash removal door passes upward through the two portions of the grate and flows through the bed of coke. The primary coking air is proportioned to the primary under grate air in such a way that a charge of fresh coal is completely coked in the time interval required to entirely burn a charge of coke. The primary coking air, primary under grate air, and secondary air are admitted to the furnace in fixed proportions at all times, and the combustion rate is controlled by a damper which regulates the effective draft at the smoke collar.

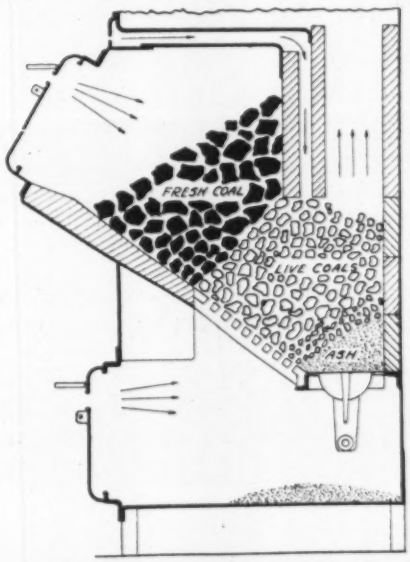

Fig. 1—Sectional view of the body of a furnace incorporating the down-draft coking principle

Methods Adopted to Improve Down-Draft Principle

The application of the down-draft principle is not required for the smokeless combustion of the coke which remains after the hydrocarbon gases have been removed. Hence for the purpose of burning the fixed carbon in the fuel, an updraft firepot is provided at the rear of the down-draft coking furnace. This arrangement makes it possible for the grate to be cooled by the primary air, as it is in any conventional updraft furnace. The grate used is made in two parts. A conventional shaking grate, used for disposing of the ash, is preceded by a sloping pin-hole grate. This arrangement makes it possible to vary the effective grate area by accumulating different amounts of ash at the back of the fire-pot. Since the forward portion of the inclined grate is always clear after a charge of fuel has been placed, the fire responds immediately regardless of the size of the coke bank. Hence, even in very mild weather, when small charges of coal are used and the fire is allowed to burn for long periods without attention, it is possible to maintain an ignition surface under the forward portion of the baffle wall.

The coking chamber at the front of the furnace is provided with a deadplate inclined toward the coke burning chamber at the back so that gravity aids in the transference of the coke from the coking chamber to the coke burning chamber. However, the angle of inclination is such that the coke does not feed on to the grate automatically but remains in the coking chamber until transferred to the coke burning chamber by poking, previous to the placing of a fresh charge of coal. With this arrange-

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ment sufficient coke for the ignition of the gases from the succeeding charge may be retained in the furnace for long periods.

Because only a small portion of the air entering the furnace is admitted through the coking chamber, it is impossible for the fuel in this chamber to become excessively hot, and the baffle wall is not subjected to extremely high temperatures. The resistance to the flow of air of the small port in the firing door is such that the resistance of the charge of fresh coal in the coking chamber is negligible, and, as a result, the rate of coking is independent of the preparation of the fuel used, providing that fine dust has been removed. The introduction of the secondary air through the baffle wall combines the advantage of cooling the baffle wall with that of preheating the secondary air.

While the coke formed in the coking chamber will bridge the space below the baffle wall, this condition causes no trouble in a down-draft coking furnace, because it does not affect the burning of the coke in the updraft fire pot below the baffle, and the various parts of the furnace are arranged so that the coke can be easily broken up and pushed into the coke burning chamber at the end of a cycle.

Clinkering of the ash is eliminated because most of the coke is burned on the inclined *pin-hole* grate which uniformly distributes the primary under grate air, and because the port which admits this air to the ash pit is designed to limit the amount at all times.

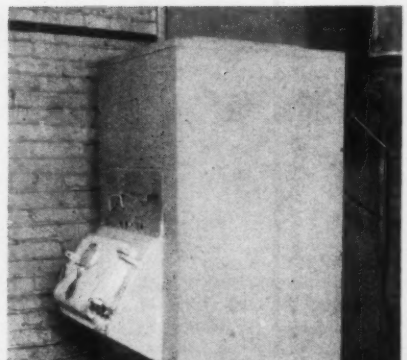

Fig. 2—Photograph of a small warm air furnace incorporating the down-draft coking principle

The tendency toward *puff-back* is reduced by using a baffle that is much shorter than those used in most of the previous applications of the down-draft principle. *Puff-back* can be definitely eliminated in a down-draft coking furnace by using the proper proportion of primary coking air.

Application to a Warm Air Furnace

Fig. 2 shows a small warm air furnace designed to operate on the down-draft coking principle, and which is now being tested in the Mechanical Engineering Laboratory at the University of Illinois. The furnace shown was developed by W. D. Redrup, Huntington, Ind., with the cooperation of Prof. A. P. Kratz, Prof. S. Konzo and the authors of this paper. This unit is 18 in. square and is fitted with a casing 26 in. by 26 in. It will be rated at approximately 60,000 Btu per hour with forced-circulation of the air. The same unit may be used with gravity-circulation, or as a circulating space heater. This furnace is so designed that it can be made in any desired capacity by maintaining approximately the same geometric proportions used in the unit now being tested.

Performance Data for a Down-Draft Coking Furnace

Some data on the performance of the original laboratory model in burning bituminous coal from 8 different fields are given in a previous[1] paper.

[1]An Improved Hand-Fired Furnace for High Volatile Coals by J. R. Fellows and J. C. Miles. Transactions ASME Vol. 64, page 161, April, 1942.

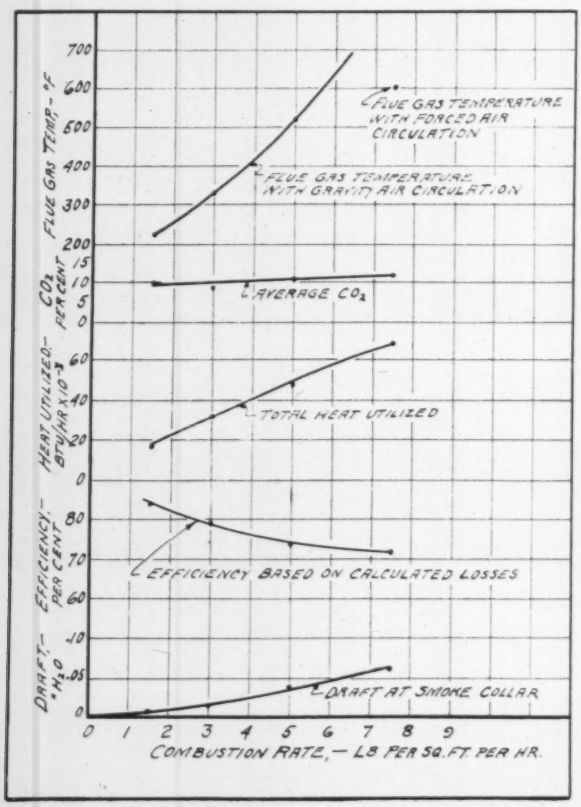
FLUE GAS TEMPERATURE WITH FORCED AIR CIRCULATION
FLUE GAS TEMPERATURE WITH GRAVITY AIR CIRCULATION
AVERAGE CO2
TOTAL HEAT UTILIZED
EFFICIENCY BASED ON CALCULATED LOSSES
DRAFT AT SMOKE COLLAR
FLUE GAS TEMP.-°F
CO2 PER CENT
HEAT UTILIZED-BTU/HR X 10-3
EFFICIENCY-PER CENT
DRAFT-"H2O
COMBUSTION RATE,—LB. PER SQ. FT. PER HR.

Fig. 3—Performance curves for a warm air furnace burning high volatile coal

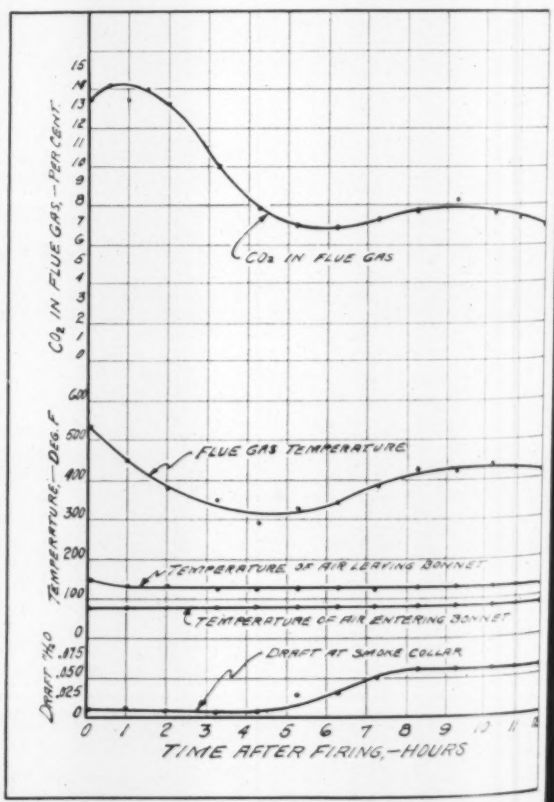

Fig. 4—Graphical log for a 12 hour cycle

The performance of the furnace shown in Fig. 2 when tested under conditions designed to simulate cold weather operation, mild weather operation and night banking, is also shown in graphical form in a recently published paper[2] Table 1 presents additional, and more recent, data taken at 4 different burning rates when burning 2 in. x 3 in. nut coal from Saline County, Ill. The proximate analysis of the coal as fired was as follows: moisture 3.5 per cent, volatile matter 38.22 per cent, fixed carbon 51.61 per cent and ash 6.67 per cent.

Fig. 3 shows in graphical form some of the more important items of the data presented in Table 1. The first 3 tests were made with gravity-circulation of the air through the casing without any attached pipes while in the last test at 7.5 lb combustion rate, approximately 700 cu ft of air was circulated through the casing per minute by means of a fan. All of the tests were made without adjustment of any of the three air dampers and in all but the 24 hour test, at a 1.5 lb average combustion rate, the cross damper in the breeching remained in one fixed position through each test period. In the 24 hour test, the cross damper was moved to a *hold-fire* position after 11.5 hours of operation at a somewhat higher combustion rate. Other tests have shown that an average combustion rate as low as 1 lb per square foot of grate area per hour can be maintained without appreciable sacrifice in efficiency and with almost 100 per cent smokeless combustion. In every case the correct position of the cross damper for the combustion rate desired for a test was determined by preliminary trials. The draft in the breeching between the cross damper and the stack was maintained constant at 0.07 in. of water in all tests. The smoke density was measured by means of a recording meter making use of a light shining through the gases on to a thermopile. The flue gas was analyzed with a standard Orsat apparatus and the temperatures recorded were measured with a thermocouple and a potentiometer. The loss of fuel through the grate was determined by weighing the refuse from the whole series of tests, and deducting the weight of ash, calculated from the weight of coal burned, and the known ash content of the coal.

[2]Performance Characteristics of a Down-Draft Coking Furnace by J. R. Fellows. Mechanical Engineering, May, 1943.

Table 1—Performance Data for a Down-Draft Coking Furnace Burning Saline County, Ill., Coal

Average combustion rate, lb per hr	1.5	3	5	7.5[a]
Length of cycle, hours	24	12	8	6
Weight of each charge fired, lb	36	36	40	45
Average draft at smoke collar, in. H_2O	0.005	0.017	0.04	0.05
Average CO_2 in flue gas, per cent	9.4	8.72	10.56	11.17
Average flue gas temperature, deg F	224	333	526	600[a]
Average heat exchanger temperature (Fire pot level)	390	613	781	725[a]
Average heat exchanger temperature (Top of combustion chamber)	324	456	726	590[a]
Average temperature of heated air	126	150	209	137[a]
Average temperature of air entering bonnet	62.3	70.0	67.5	64
Average smoke density, per cent (Corrected for zero reading of recorder)	0.6	2.0	4.0	1.0
Heating value of coal as fired, Btu per lb	12,880	12,880	12,880	12,880
Total stack losses, Btu per lb of fuel	1,217	1,748	2,458	2,810
Grate loss, Btu per lb of fuel	847	847	847	847
Total heat utilized, Btu per lb of fuel	10,731	10,285	9,475	9,223
Total heat utilized, Btu per hour	16,224	30,855	48,375	69,172[a]
Overall efficiency, per cent	84	79.8	73.6	71.6

[a]Forced circulation of air being heated.

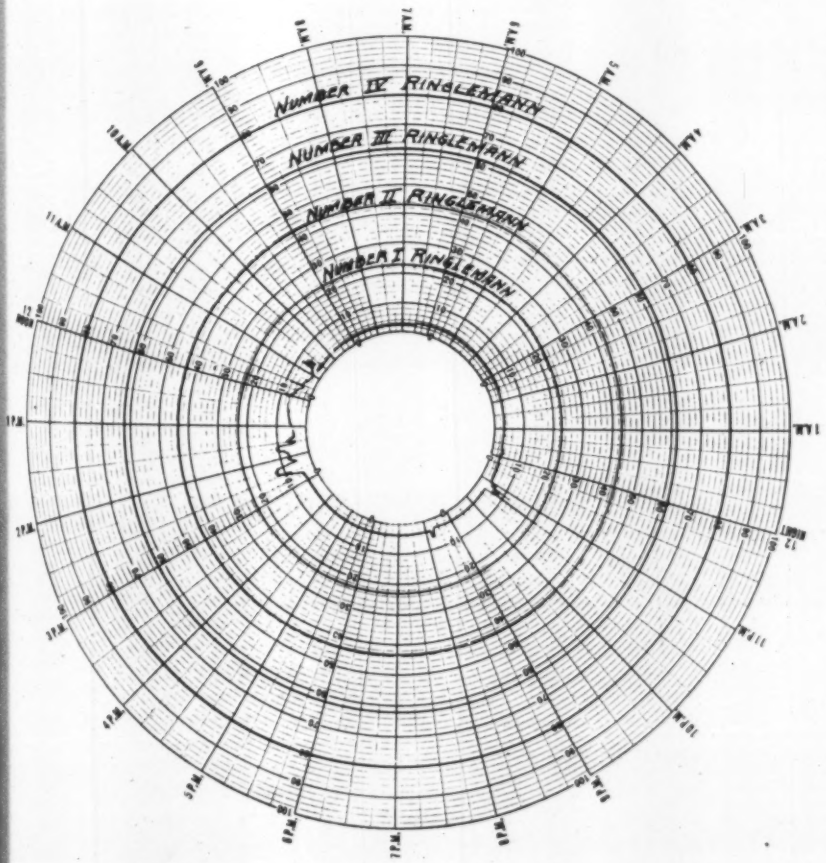

Fig. 5—Reproduction of an actual smoke record

It may be noted that the average smoke density in all of the tests was negligible when it is considered that a density of 23 per cent as recorded by the meter corresponds to a Ringelmann number of 1. The smoke density did not at any time reach a density greater than 15 per cent as measured by the recording meter when corrected for the zero reading of the instrument.

It may also be noted from the table, that, in spite of the fact that the fuel bed received no attention whatever except when each charge was placed, the furnace operated with a good overall efficiency at all of the different rates of burning. It is believed that the grate loss, which was quite high, can be greatly reduced by a slight change in the position of the horizontal rocking grate, as in its present position most of the ash is shaken through at the front edge, whereas it should be arranged so that the greater part of the ash is shaken through at the rear edge next to the rear wall of the furnace.

Fig. 4 shows a graphical log of the CO_2, the flue gas temperature, the temperatures of the air entering and leaving the casing, and the draft used, in a 12 hour test in which the position of the cross damper was occasionally changed as much as was necessary to maintain a constant heat output of approximately 35,000 Btu per hour from the casing. The coal used in this test was a 2 in. by 3 in. preparation from Franklin County, Ill. Since it was possible to maintain a continuous output equal to more than half of the rating of the furnace for a period of 12 hours, the test shown in Fig. 4 indicates that a furnace of this type would not need to be fired more often than twice a day except in unusually

cold weather. Except for a short period following the placing of a charge of fresh coal, the heat output of the furnace was readily controlled by adjusting the position of the cross damper. The fact that all of the available draft was not being used when the heat output was low indicates that a thermostatic control of the damper could have maintained a uniform heat output throughout the cycle, except at the beginning. A furnace of this type could be operated with thermostatic control immediately after placing a charge of fresh coal without fear of explosion or *puff-back*. It is probable that some over-heating at the beginning of the cycle might be experienced because of the high combustion rate developed during the few minutes that the firing door was open in the process of preparing the fuel bed and placing the coal. However, it does not appear that the overheating would be sufficient to cause discomfort, and it is believed that the rapid response of the furnace after placing a charge is a very desirable performance characteristic. The necessity for overcoming the phenomena of *puff-back* is principally responsible for the fact that the furnace did not maintain high CO_2 throughout the whole cycle. Sufficient primary coking air must be supplied to carry away the gases released in the coking process during the time when that portion of the charge which is close to the firing door is being coked. The minimum amount of coking air required to eliminate *puff-back* determines the adjustment of the coking air damper, and it is impractical to attempt to extend the coking of a charge of fresh coal over the duration of a complete firing cycle. The requirements for secondary air are therefore considerably greater during the first part of each cycle, and it is impossible with one setting of the secondary air damper both to provide a sufficient amount at that time and to avoid an excess during the later part. Tests have shown that, by careful adjustment of the three air dampers, CO_2 readings as high as 17 per cent can be obtained without smoke. Covering each charge with fine coal reduces the tendency toward *puff-back*. This makes it possible to reduce the amount of secondary air used, and effects a corresponding increase in the average CO_2. More frequent firing of smaller charges of coal would result in a higher CO_2 average. Reducing the amount of secondary air at the midpoint of each cycle would also have the same effect.

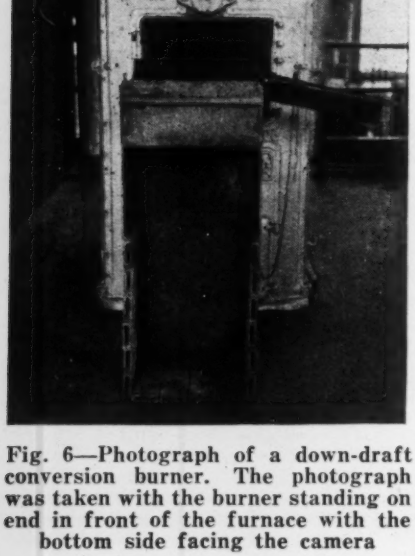

Fig. 6—Photograph of a down-draft conversion burner. The photograph was taken with the burner standing on end in front of the furnace with the bottom side facing the camera

Fig. 5 is a reproduction of the actual smoke record made at the time the data shown in Fig. 4 were recorded. It also includes the record for the following 12 hour cycle, for which the same coal preparation was used. At the time this record was made, the recorder was adjusted to a zero reading of 4 per cent, so that it is only readings in excess of that amount that indicate other than a perfectly clear stack. It is possible to burn all types of high volatile bituminous coal in a down-draft coking furnace without producing any smoke whatever but any one of a number of unusual situations may cause a light smoke at times. The trace of smoke which appeared at times during the fore part of the test cycle was probably caused by the fact that the accidental positions of the various individual pieces of coal making up the charge were such that an uneven distribution of the gases passing under the baffle wall resulted. Thus, while sufficient secondary air entered the furnace and was distributed uniformly across the lower face of the baffle, it is possible that there was a slight deficiency at one or more points. The use of a smaller sized coal reduces the chance of unequal mixing of the secondary air and coal gases, and increases the chance for obtaining 100 per cent smokeless combustion. It may be noted that no smoke was produced in the second cycle shown in Fig. 5 except for a few minutes immediately after firing. The small amount of smoke produced at the beginning of this cycle was caused by suddenly checking the fire subsequent to the very high combustion rate that was built up while the firing door was open and the fuel bed was being prepared and the charge placed. However, if the position of the Ringelmann number 1, obtained from a calibration of the recorder, is noted, it becomes evident that the amount of smoke produced in the 24 hours of operation was negligible. Allowing the fuel bed to burn too low before refueling, a situation apt to occur only in very mild weather, will of course make it impossible for the operator of a down-draft coking furnace to avoid the production of a light smoke for a more extended period. However, if the fore part of the inclined pin-hole grate is properly cleared of ash, a hot flame is soon established below the baffle wall and the smoke produced is reduced to a minimum.

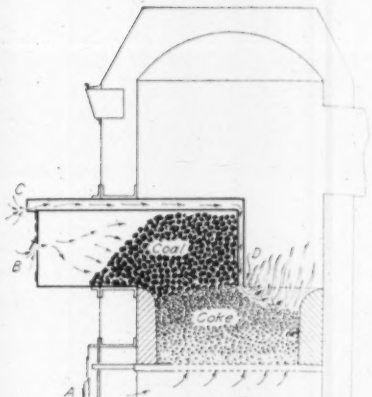

Fig. 7—Vertical cross-section of test furnace with radiator removed and with down-draft burner in place

Down-Draft Conversion Burner

The down-draft coking principle was first applied to conventional up-draft furnaces by means of the Azbe baffle developed by Victor J. Azbe, St. Louis, Mo., about 1926. The results of tests with the Azbe baffle made under Mr. Azbe's direction by Prof. P. F. Mohn at the Firing School in St. Louis during the summers of 1927 and 1928 suggested the idea of the down-draft conversion burner to the senior author in January 1936. As a result of a large number of tests in more than a score of home furnaces and the construction of nearly sixty experimental models, it is believed that a practical design has been developed. Fig. 6 is a photograph of the burner used in numerous tests[3] made in the Mechanical Engineering Laboratory at the University of Illinois. Fig. 7 shows a longitudinal vertical section of a warm air furnace with a burner in place. Fig. 7 shows the condition of the fire immediately after placing a charge of fresh coal. The principle of operation is exactly the same as that of the down-draft coking furnace. Each charge of fresh coal is coked in the burner while the coke from the preceding charge is burned in the fire pot of the furnace. The secondary air enters the burner through a slot above the firing door, passes through a horizontal air passage in the hollow roof, and then through vertical passages in the rear and side walls. It then emerges at the bottom of the walls where it mixes with the gases from the coking coal. The gas-air mixture is ignited by coming in contact with the incandescent coke from the previous charge as in the down-draft coking furnace. A down-draft conversion burner can be installed in a conventional furnace in a few minutes without making any

[3]Burning Illinois Coal Smokelessly in Hand-Fired Heating Plants by J. R. Fellows and J. C. Miles. Circular No. 43, Engineering Experiment Station, University of Illinois.

changes in the furnace other than the insertion of the burner through the firing door opening. After the installation of the burner a conventional furnace possesses all of the advantages of the down-draft coking furnace except for the absence of the inclined pin hole grate.

During the winter of 1941-42 a down-draft conversion burner was installed in a warm air furnace heating a two story wooden building having a calculated heat loss of 346,000 Btu per hour based on an assumed outdoor temperature of 0 deg. F. A record was kept of the fuel consumed in the furnace and of the temperature maintained in the living quarters of the structure. Simultaneous fuel consumption and temperature records were also kept for an identical building heated by an identical furnace fired by the same fireman from the same pile of Jackson County, Illinois coal. Records were kept for a six weeks period covering almost a complete range of winter weather conditions. The results proved that less fuel was burned in the burner equipped furnace in every type of weather experienced during the test, and that the average saving in fuel effected by the burner was 24.1 per cent, based on the weight of fuel burned in the other furnace. The temperature in the living quarters was much more uniform in the building that was heated with the burner equipped furnace, and though there was far less overheating the average temperature exceeded by 3.8 degrees, the average temperature maintained by the furnace that was used in the conventional manner.

Other Possible Applications of the Down-Draft Coking Principle

Since the down-draft coking principle applies only to the method of burning the fuel and is not affected by the design of the heating surface for utilizing the heat liberated, it is believed that it can be applied to stoves, boilers and water heaters as easily as to warm air furnaces. An attempt is now being made to apply the principle to a simple space heating stove. The authors believe that the down-draft coking principle offers the closest possible approach to 100 per cent smokeless combustion of high volatile fuels in hand-fired heating appliances, and an attempt will be made to develop all of the many possible applications in the near future.

Appendix

Kindling a Fire. The correct procedure to use in starting a fire in a cold furnace of the down-draft coking type is to clear the greater part of the inclined pin-hole grate, fill the fire pot with coal to a level 3 to 4 in. below the bottom of the baffle, place several short sticks of dry wood on top of the coal and ignite the wood with a few pieces of crumpled paper. In 3 or 4 hours, after the starting charge is thoroughly coked, the coke in the fire pot may be broken up and a charge of fresh coal placed in the coking chamber. It is easier to start a fire in a down-draft coking furnace than in a conventional furnace because the fire pot is more accessible from the firing door opening, and while it is impossible to avoid the production of a light smoke from the starting charge, the coal is ignited at the top and burns from the top down with plenty of secondary air supplied directly over the surface and, as a result, the density of the smoke produced by the starting charge is much lower than the minimum density produced in kindling a fire in a conventional furnace.

Discussion

W. M. Myler, Jr., Columbus, Ohio, (Written): It is very gratifying to know that the Experiment Station of the University of Illinois has been able to successfully apply the down-draft coking principle to a hand-fired domestic furnace which is simple enough in operation that it has a reasonable chance of performing satisfactorily in the hands of the average housewife.

It is particularly gratifying that this system of combustion can be accomplished satisfactorily by means of a conversion burner which can be installed in an average furnace. If this were not possible, the use of this method of combustion would be postponed indefinitely.

In normal peace times, the smokeless feature of the operation would undoubtedly be the one which would command primary interest. At the present time, when transportation facilities are strained to the utmost in being able to supply fuel for all purposes, anything that would substantially increase the fuel economy of a domestic hand-fired furnace should be given whole hearted endorsement. It is, of course, too good to hope that in the hands of the average housewife that as high efficiency would be obtained as is reported in these tests. If we consider, however, that the average housewife with her present equipment is attaining probably only a maximum of 50 per cent overall efficiency, and that with this new equipment it appears to be possible for her to obtain half the possible increase, thus permitting her to obtain 60 per cent efficiency, this would represent a fuel saving of approximately 16⅔ per cent, which is certainiy a well worthwhile saving. In addition the smokeless benefits would be achieved at the same time.

It would seem that with the prospects of such a large increase in efficiency within the range of probability, it might be possible to get the government powers, having jurisdiction, to allocate sufficient metal to make up conversion burners, at least in limited quantities, to take advantage of this fuel saving.

In reviewing the data presented in Table 1, the question has arisen in my mind regarding the item *grate loss, Btu per pound of fuel,* which is shown as 847 Btu per pound of fuel for all rates of combustion. In view of the fact that the weight of ash was determined as a total for the entire series of tests, this is the only way that this could be recorded. It has been the writer's past experience, however, that the grate loss is not always the same for all rates of combustion. Quite frequently at the lower rates of combustion, more unburned carbon will find its way into the ash, so that it would seem that the reported results may show too high an efficiency for the lower rates of combustion and probably not quite sufficiently high efficiency for the two higher rates of combustion.

The effect of such a change would be, of course, to flatten out the efficiency curve and actually give an increase in efficiency at the higher rates which are the rates at which the furnace would be operating during a considerable portion of the time and perhaps at which it would show a larger saving of fuel.

C. F. Newport, Chicago, Ill. (Written): A hand-fired boiler or furnace that will warm the home to a desired temperature for long periods without attention is most desirable. Long firing periods can be had by burning anthracite; and long coaling periods with even room temperatures under varying weather conditions can be had when burning the smaller sizes of anthracite in a magazine feed type boiler.

A long firing period with bituminous coal is more difficult to obtain. The authors seem to have achieved long firing periods when burning high volatile bituminous coal by coking the coal before burning it and thus eliminating objectionable smoke. Instead of feeding the coal to the fire by gravity from a storage magazine (which is a difficult proposition with soft coal) they feed the coal to the fire by hand after it has been coked in a coking chamber that holds enough coal for one firing period or cycle.

The draft pull for the furnace is held constant (apparently by a barometric draft control) and the heat output is controlled by a choke damper close to the furnace. It seems doubtful if this damper can be controlled by an ordinary electric thermostat and damper motor because of its off and on positions, but the ideal automatic control would be one obtaining a gradual control by varying the position of the damper to suit the room temperature requirements.

Will gases escape through the air inlet to the coking chamber or the air inlet to the baffle, if the choke damper is closed enough to hold the fire in check, when little or practically no heat output is required?

Since air entering the furnace through the three air inlet ports must be proportioned correctly for proper results it seems that the correct setting of each of these ports will require the services of an expert. Even though they were set at the factory it would still be necessary to adjust the barometric control in the smoke pipe to maintain a constant draft pull through the furnace. These conditions can be obtained in a laboratory, but for successful operation similar conditions must be obtained in the field, where varying conditions of installation prevail.

Will tar and condensed moisture form on the walls and door inside the coking chamber, when burning certain types of coal?

As the method of firing is different from that with which the ordinary home owner has become familiar with he will very likely require special instructions on how to fire for proper results. Although the performance of the furnace has shown such good results from the tests, it remains to be seen how it will perform in the field.

R. A. Sherman, Columbus, Ohio (Written): The commendation which it is customary for a discussor to extend in his opening remarks has been well earned by the authors, as the senior author has spent seven years to develop the furnace described. However, in consideration of the difficulties of the task of accomplishing smokeless combustion in a non-mechanical device and the many man-years of effort that have previously been expended on the task, this price is not great.

The opening statement of the paper that the temperature required to ignite a coal gas-air mixture is much higher than that required to release the volatile matter from the coal is somewhat exaggerated. Although it is true that volatile starts to be driven from coal at 700 to 900 F, the range in which it is rapidly released is in the same range as the ignition temperature of the gases, that is, 1100-1200 F.

The difficulty from smoke in hand-fired furnaces is usually not that a high enough temperature does not exist at some place in the furnace for the ignition of the volatile, but that this temperature is not maintained in the zone where the volatile can meet air for its combustion.

Control of the rate of liberation of the volatile matter to match the rate of admission of secondary air is essential for smokeless combustion. The authors control that rate by the rate of admission of coking air through the charge of coal. It is interesting that they have found it possible to place the control of the rate in the size of the orifice through which the air is admitted and independent of the resistance of the fuel bed. Data on the pressures in the various parts of the furnace during a complete cycle would be a valuable addition to the paper.

The efficiencies reported in Table 1, and in Fig. 3, are surprisingly high, and it is to be regretted that the heat balance data were not more completely given. The calculation of the ash pit loss as uniform for all rates is not justified but probably introduces no great error. No data are given for the loss in CO and it is assumed that they are included in the stack losses or that there were none.

The heat output and efficiencies given are obtained by difference between the heat input and the sum of the stack and ash pit losses. They thus include the radiation and any unmeasured losses. The values would be more convincing if at least one directly measured efficiency were given or if a test had been run with coke or anthracite with which the unaccounted losses are all very low.

To gain some conception of the agreement between the directly and indirectly measured efficiencies, the writer calculated the output in the high rate test reports in Table 1. The output was calculated from the value of 700 cfm, given as the approximate rate of air circulation, and the rise in temperature. This gave a value of 55,188 Btu per hour for the output and 57.1 per cent for the efficiency as compared to the values of 69,172 and 71.6 per cent which the authors calculated by difference. It is possible, of course, that the air flow was in error by that amount.

It is to be hoped that the authors will obtain further confirming data on the directly-measured output and efficiency in their future tests. High efficiency is a desirable feature, although possibly less essential than smokeless operation, but if the latter is obtained by complete combustion of the volatile hydrocarbons normally lost, high efficiency will naturally follow.

The furnace is not foolproof for, as the authors point out, it is essential that it be fired at proper intervals, but this is surely not too much to ask of the users if they are to receive the benefits described.

M. W. McRae, Chicago, Ill., asked the authors if they had any difficulty with the stationary grate burning out due to most of the ash being deposited on the shaking grate, and also if any trouble had been experienced with breakage of the down-draft ducts, which he understood were made of refractory.

Professor Fellows: I shall consider the questions as presented. (1) There was a question about smoke backing out of the door. I am not sure whether the questioner had in mind the firing door or the secondary air ports. The authors have not experienced the slightest tendency toward smoke backing out through either of those openings. There is a very strong draft effect in those small vertical flues, in which the burning takes place. It has been the authors' experience that the furnace in the laboratory will operate successfully on a draft at the smoke collar as low as one-thousandth of an inch of water. As a matter of fact, the authors have actually operated, just as a stunt, with a slight positive pressure at the smoke collar. That can only be made possible by the phenomenon known as static pressure regain. As the gases pass through the radiator toward the smoke collar, the temperature of the gases decreases, the specific volume correspondingly decreases, the velocity decreases, and there is a chance for conversion of kinetic energy to static pressure. Operation has been for short periods with positive pressures as high as five thousandths of an inch of water at the smoke collar. The authors can say definitely that they do not believe there will be any difficulty from that source.

(2) Did soot and tar gather in the coking chamber? There is considerable accumulation of soot, but we have not noticed any accumulation of liquid tar on the side of the coking chamber. There is, we believe, no more accumulation of soot on the inside of the firing door than in the conventional furnace.

(3) Will special firing instructions be required? The technique of firing the furnace is extremely simple. It is important, of course, that the proper technique be followed. The technique consists simply of shaking the grate and pushing the coke back. The only rule is, just push the coke back as far as it can go. It pushes back very easily because of the shape of the floor of the coking chamber and the inclined pinhole grate. The authors have talked with a number of furnace salesmen and they seem to feel that it will not be difficult to instruct the public. In the first place, the operating instructions will be very simple; and in the second place, it is something different. The public will not expect to know how to operate it without instructions and we believe that they will follow the simple instructions that will be furnished.

(4) Will the draft control be effective? It will be necessary, as in any hand-fired furnace, for the operator to use judgment in firing the furnace in very mild weather during the early fall and late spring. He will experience overheating if he puts in a charge of coal that would be suitable for cold weather. The proper procedure to use in mild weather is to fire a small charge of coal once every 24 hours, putting in the coal in the evening, because invariably the nights are cooler than the days. The volatile matter will then be burned off through the night when there is more demand for heat and the coke will hold fire through the day without the production of much heat.

(5) Should grate loss be the same for all rates of combustion? The authors thoroughly agree on the variation of grate loss with different combustion rates, but they had no other way of determining the grate loss, as there was not sufficient time available for running long tests at each combustion rate.

The authors disagree with Mr. Sherman's comment, that the opening statement of the paper is exaggerated. S. W. Parr and H. L. Olin reached the following conclusion after exhaustive studies in the coking of Illinois coal[4]: Coals of the Illinois type can be coked at a temperature approximately 400 deg to 450 C (450 deg C corresponds to 842 deg F). J. W. McDavid[5] concluded that a temperature of 1500 F is required to insure the instantaneous ignition of a coal-gas-air mixture when it is heated by the sudden application of a hot body.

Parr and Olin's experiments indicate that all of the smoke-forming hydrocarbons in Illinois coal are converted to the gaseous state at temperatures below 842 F, whereas McDavid's experiments indicate that a temperature of 1500 F is required to insure the ignition of all of them.

If smokeless combustion is to be achieved, all smoke-forming gases must be mixed with air and the mixture must be forced to pass some point in the furnace where the temperature is high enough to insure its ignition.

The authors agree with Mr. Sherman that further tests to determine

[4]The Coking of Coal at Low Temperatures, University of Illinois Engineering Experiment Station Bulletin No. 60.

[5]The Temperature of Ignition of Gaseous Mixtures, Transactions, Chemical Society (London), Vol. 3, 1917, pp. 1003-1015.

the bonnet efficiency should be made at the first opportunity. While it is essential that the furnace be re-fired after the completion of the coking process and before the greater portion of the coke is consumed, this interval during which the furnace may be properly attended would continue for several hours under the operating conditions experienced in normal winter weather.

Concerning air distribution and control we refer those interested to a paper[6] recently published, in which it is pointed out that orifice size rather than fuel bed resistance determines the air distribution.

The authors wish to express their appreciation to those who have taken the time to carefully read this paper and prepare written comments.

Mr. McRae questioned about the stationary grate burning out. This grate was carefully examined just before leaving Urbana, scraping it clean and examining it from both sides after approximately 5 months of operation, part of the time at maximum combustion rate. Not the slightest tendency toward distortion or oxidation was detected. Of course the burning coke does lie directly on the grate and the grate is not protected by ash; however, some air is continually passing through the grate and the cooling seems to be sufficient to prevent overheating. The pinhole construction prevents localized areas of intensive burning and this is a help in preventing damage to the grate. This arrangement also reduces the tendency toward formation of clinker. As a matter of fact, it definitely eliminates the formation of clinker. We have not had any tendency toward clinker formation with any types of coal used, and some of the coals used have been coals like catlin coal, which has a very strong tendency toward clinkering. In other words, it has an ash with a very low fusion temperature.

Mr. McRae also questioned the downdraft refractory flues. The authors have not had as much experience with that construction for the baffle wall as they would have liked, but no difficulty was experienced up to this time. The refractory problem was discussed at length with officials of a large refractory company. The furnace in its present stage is purely and simply a refractory problem, and its success depends on whether or not any difficulty with the refractory arises. The authors have been experimenting with refractory shapes in the construction of the baffle wall for about two years, and they have had no difficulty with any of the shapes used. Unfortunately, the authors have not had a long experience with the particular design that is being used right now.

[6] An Improved Hand-fired Furnace for High-volatile Coals, ASME Transactions, April 1942.

REME ORGANIZED

To meet a long felt need of the British Army a new corps, the Royal Electrical and Mechanical Engineers, was officially organized on October 1, 1942 for the purpose of conserving the nation's technical manpower and to coordinate the electrical and mechanical military engineers into one body. The REME embraces almost the entire range of mechanical fields and the functions of the Corps are briefly as follows:

1. Inspection and maintenance of tanks, wheeled vehicles, all artillery (including field, anti-aircraft, and coast defense) small arms and machine guns, radiolocation, fire control, and all other instruments, tunnelling equipment, pumping sets, and the installation of coast artillery machinery.
2. Repair of all the above equipments consequent upon ordinary wear and tear, or battle casualties.
3. Investigation into defects of design and recommendations for improvements.
4. Advice on prototype design from a maintenance angle.

The announcement of the authorization and the formation of this new branch is given in the Journal of the Institution of Mechanical Engineers (September and November 1942).

In a talk before a meeting of the Institution on February 12, 1943, Col. R. B. Maxwell, Assistant Adjutant General, REME, described the formation of the Corps, the field organization, the need for engineers in large numbers for the fighting services, and the industries behind the lines.

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A Forum on the Drying of Foods*

History and Present Scope of Dehydration

By Morilla Carveth,† Chicago, Ill.

DEHYDRATION OF food products, as practiced today, is really drying with a college education.

The drying or dehydration of vegetables is the oldest method of food preservation known to man. Dried foods found in the ancient tombs of Egypt in the *City of the Dead* indicate the sun drying or dehydration of some foods was an art practiced over 9000 years ago.

Every major conflict has brought the need for foods preserved in this manner to the attention of humanity. Foods were dehydrated during the Boer, Civil, and Spanish American wars. Most of this material was of such unsatisfactory quality the activity was discontinued as soon as the pressing need was over. Little attention was paid to nutritional or keeping qualities. Research and scientific control had not entered the picture.

Some few spartan souls continued with experimental work. They had faith that we might be destined to find the answer to the problem of food shortages which nature and catastrophe had been hurling at mankind since the *seven years of plenty* were followed by the *seven years of famine* in Egypt.

Dean Prescot of Massachusetts Institute of Technology, Dr. Cruess at the University of California, and Dr. Tressler of the New York Experiment Station, and others, continued to carry on some experimental work, but by and large interest in dehydration became almost nil during the 15 years prior to 1941. In 1941 there were nine small vegetable dehydration plants in operation in this country. Their total capacity was less than 9,000,000 lb of finished product. Dehydrated soups had been established on the commercial market. Irish potatoes, dried as shreds and in julienne strips, had been successfully marketed, and existing facilities were never able to supply the demand created for onion flakes and powder, garlic, celery, and parsley. The dried egg industry had committed suicide in 1917, along with the vegetable drying group, but they too had reestablished themselves on a quality basis. The baking and confectionery industries, and institutions, had found both dried egg and milk satisfactory products for their use. Two large egg producers sold several carloads of their product in small packages on the domestic market. Their survey of sales clearly indicated the domestic commercial market was ready for both dried eggs and milk powder. Potato shreds, onion flakes, parsley, and garlic had been readily accepted by the American housewife, and some distribution of these products had been made through local grocers. (Average retail price per package was 15 cents.)

Dehydration, as an industry, was expanding slowly in this country. During the same period, it was growing by leaps and bounds in Europe. In 1938, Germany dehydrated more potatoes than were grown in the United States. Over 1000 plants were in operation, and through such operation, they had learned the necessity for scientific control. In 1941, the lack of progress in dehydration was realized. The personnel of the nine small plants were called together and the situation discussed. The Army wanted 25,000,000 lb of these products, and industrial institutions had been slowly learning to appreciate—25,000,000 lb, and the capacity of the plants in operation was less than 9,000,000.

The intensive program inaugurated to satisfy the demand has brought growing pains never equalled in industry. Much has been learned from these pains.

The old natural draft dryers have been found unsatisfactory for present operations. Some forced draft tunnel dryers previously operated on fruits in the West Coast area have been converted to vegetable operations with varying degrees of success.

The No. 1 improvement in the process is blanching. Blanching has been adopted as part of the processing of foods because it inhibits or renders inactive certain chemical components or enzyme systems naturally present in the tissues of the plant.

The No. 2 improvement in the industry has been the reduction in moisture content in the finished product. In 1917 and 1918, the government specified a moisture content of 10 per cent in the products manufactured. We have since learned that keeping qualities of dehydrated products are immeasurably improved by reducing the moisture content to 5 per cent. It may well be that 5 per cent is a compromise with quality because of the necessity to convert and utilize existing facilities.

No. 3 is the utilization of the knowledge research has developed of the products themselves. No longer does one try to dehydrate the quick growing, succulent varieties because cell structure has been studied. The physical and chemical conditions to which raw materials are subjected has been related to their known susceptibilities. Mechanical variables and such physical variables as variety and age of the product are given consideration. New plant hormones and other biologically active materials are being used in plant breeding. Controlled fertilization and irrigation are increasing nutritive values.

Today the dehydrated food desired by the Army is limited only by the amounts obtainable. The 1943 requirements are now estimated at 1,750,000,000 lb. The vegetable portion is placed at 400,000,000 lb of this amount.

No. 4 is rapid preparation. Rapid preparation is essential for vitamin retention. The advantages gained through improved cultural practices and rapid harvests can all be lost by slow, improper preparation and drying. Due consideration must be given the natural characteristics of the products.

No. 5. Today there are many designs in drying units. The majority of these designs are of the tunnel type. They are designated as counter, parallel, cross flow, or center exhaust tunnels. Their names are descriptive. The center exhaust is a combination of the counter and parallel flow units. Cabinet or batch drying units are used successfully in smaller operations. Fluid products are being handled in spray dryers. The use of vacuum units is limited because of the critical material involved in their construction, but it is evident that they will play an important part in the dehydration picture. There are exact physical laws which describe accurately the operation of every dehydrator. These laws may be complex, but there is no mystery about the conservation of mass and of energy and the second law of thermodynamics, which rule in all physical processes.

Much additional research is needed. We must learn more about enzymes—their inactivation in preparation, and their importance in determining keeping qualities. Preparation and drying time must be shortened. The American housewife will demand (1) high

*Three papers presented before the Illinois Chapter at its meeting on April 12, 1943, at the Merchants and Manufacturers Club, Chicago, Ill.

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have, they have perfected a practical mass production unit known as the Eidt tunnel which has been copied in this country with some variations. The fundamental principle of this tunnel is that the product is exposed to high temperature air at the beginning of the process, air of lower temperature as the drying progresses to avoid scorching. However, as dry air as possible is used at the finish so drying will be complete.

In order to accomplish this, air is introduced from both ends of the tunnel and flows parallel with the product during the start of the process, and counter current during the last portion. The air is exhausted near the center and for this reason, this tunnel has been called the "center exhaust" in this country.

Provisions are made for recirculating the air to the most economical extent. There are two entirely separate fan systems with heating coils, dampers, and automatic control valves for temperature regulation. The controls are usually mounted so the temperatures can be regulated at two or more places. An average tunnel of this type is about 90 ft long, holding 15 trucks, with trays about 8 ft long and 4 ft wide. This will handle about 20 tons of raw product per day, giving about 3 to 3½ tons dried potatoes. These tunnels are usually built in the field under a construction contract. Another variation of this is the so-called multiple tunnel which consists of one parallel flow, used as a predryer, and two counter flow tunnels used as finishing tunnels. This uses the best features of each type and may have some possibilities.

The transverse flow tunnel is one in which the air flow is transverse rather than longitudinal. In the better tunnels of this type, provision is made for discharging a measured quantity of air directly across each tray to insure uniform distribution. These tunnels are usually short, having at least two fan systems with independent outside and exhaust air connections. This permits different temperatures to be carried in each section. The principal advantage of this type is that the air passes over each tray only once before it is reheated and dry air added. In other words, the air gets a shot at only one tray before it is reconditioned. This arrangement makes it possible to use the highest temperature air at every tray and at each phase in the drying. It also gives very good control over each section since their operations are independent of each other. Air is introduced over the trays through an injector type nozzle. The amount of air handled in one of these dryers is usually large and this is considerably increased due to the induced air flow made possible by the nozzle design. These tunnels are a rather recent development, but they have been thoroughly tested in actual operation.

One trend in drying interests me very much. While drying vegetables, it has been noticed that most of the moisture is given up very rapidly in the first half of the process. Some tests on white potatoes were run recently. Moisture samples were pulled at 2 hours and were found to be approximately 15 per cent. The next 2½ hours were spent reducing the moisture content to below 7 per cent. Obviously, this is a great waste of time and tray space in the dehydrator. This has given rise to the finishing bin or equalizing bin. These are usually nothing but a v-shaped bin into which the partially dry product is emptied. A relatively small amount of air is forced directly through the product. This air is kept below the critical temperature. Using a finishing bin permits a shorter drying schedule in the dehydrator and also equalizes the moisture content of the entire charge.

On some products, notably onions, due to their low critical temperature, it is necessary to use artificially dried air in the finishing bin in order to dry them completely. Experiments have been carried out with silica gel and refrigeration, and proven very satisfactory. In some localities where a high wet-bulb is prevalent, such equipment will be a necessity and the entire packaging room may have to be air conditioned. This, of course, depends on the locality and the product as some vegetables are much more hygroscopic than others.

Heating and ventilating engineers can contribute a good deal to this industry because of their knowledge of air and heating equipment. Air flow is still a major problem in most dehydrators, and could stand quite a bit of renovating. The steam piping in some plants is a disgrace. Little or no attention has been paid to normal practices and, consequently, the operation is very sluggish. One thing to remember in dehydrator design is that while all vegetables dry somewhat the same, a different schedule, as to time and temperature, must be worked out for each different type of vegetable and for each specie. This can only be done by actual experiment. In the event you are called upon to design a dehydrator, much actual experimental work must be done first in order to be sure, and then when you have this work finished, you had better ask yourself how good a gambler you are before you go ahead.

The Spray Method

(Liquid Drying)

By Samuel Van Deest,†† Chicago, Ill.

DRYING IS a very ancient art, but one for which the amount of accurate information is quite meager. It may be divided into:

1. Drying of solid material, i.e., material in which the solid phase predominates and the moisture present is there as capillary moisture.

2. Drying of liquids, i.e., material which is in solution or suspension. This is known as "evaporation to dryness" in laboratory procedure and is but one phase of the general subject of evaporation.

Fundamental Theory

The process of evaporation depends on the part that liquids exhibit, a phenomenon known as *vapor pressure*.

Ways of increasing the amount of evaporation from solutions immediately suggest themselves:

1. Heat the solution (increase its vapor pressure).

2. Heat the air it is exposed to (decrease its vapor pressure).

3. Remove the air as fast as it becomes saturated.

††Production Specialist, The Douthitt Corp.

A fourth way, not so obvious, is to increase the surface of the liquid exposed to the air. Evaporation is a molecular phenomenon and takes place because molecules of the liquid escape from the surface. Evidently if the surface of the liquid is doubled, twice as many molecules of liquid may escape as before. From this and the first two ways described, the general law of evaporation of liquids may be put thus:

The rate of evaporation of a liquid is proportional to the surface exposed and to the difference in vapor pressure of the liquid and of the medium to which it is exposed.

The medium usually used in spray drying is air. It is heated by steam coils or other means and blown into the drying chamber.

The specific heat of air is low (0.2375 as compared to 1 for water); consequently, large volumes of air are necessary. The air serves the purpose of furnishing some of the heat for evaporating the water and also as a means of carrying the water away. Some of the heat necessary to do the evaporating comes from within the product sprayed. The more heat we are able to put into the product itself before spraying without damag-

nutrition; (2) palatability; (3) appearance and texture; (4) ease of preparation or reconstitution; (5) low cost; and (6) excellent keeping qualities. With these six qualifications, dehydrated food will take its place alongside the other established types of food—fresh, canned, and the newer frozen products. The ultimate scope of the industry depends partly on the duration of the war. The investment in dehydration plants now in operation will vary from $50,000 to $300,000. Two hundred of them will be in operation in the near future. Plant management is going to protect that investment. Some of the finest industrial scientific brains of the country are working to improve the product and reduce the cost.

Surely, the possibilities of an industry with these requirements is a real and practical challenge to the engineering brains of this country. May they do as well in meeting it as they have in the older canning industries, and in the relatively young frozen food industry.

Tunnel Dehydration

(Vegetable Drying)

By David A. Forberg,** Chicago, Ill.

BEFORE WE get into the discussion of tunnel dehydration, a little time should be spent following the product from the field up to the point of drying. This example is going to deal with white potatoes. As these potatoes come from the field or the storage bin, they are usually put through a washer to remove the field dirt and then graded as to size. This is followed by mechanical peeling which may consist of one of several types. There is the abrasive peeler —either batch or continuous—lye peeling, or the less common flame peeler. The flame peeler consists of a hot chamber through which the potatoes pass. The temperature in here is around 2000 F. This chars the skin of the potato and a subsequent high pressure wash removes most of the burned skin. Following the peeling, the potatoes go to a trimming table where any bad spots are manually removed. The potatoes are then put through the cutters, which may either be a dicer or a stripper. This is usually followed by a light wash and the potatoes are then blanched in live steam to inactivate the enzymes. Another wash follows the blanching to remove the starch and the potatoes are then ready to be loaded on trays. This preparatory schedule varies for each vegetable, but usually follows the same general procedure. The success or failure of a dehydration project is more often than not found in this phase, as no dehydrator will turn out a better product than is put into it.

The dehydrator is the means of removing moisture from the prepared food. Present specifications call for a moisture content of not exceeding 5 per cent for the average vegetable. This goes as low as 4 per cent for cabbage and as high as 7 per cent for potatoes.

This presents quite a problem if we are going to do it in a reasonable length of time and still get the production that is required. The idea is to dry these vegetables as quickly as possible without injuring them. During the first stage of drying, a large amount of surface moisture will usually protect the product from scorching. During this time, the temperature of the product itself is approaching the wet-bulb temperature of the air. As the product dries and the surface moisture disappears, two things happen. The effect of evaporative cooling diminishes and the product temperature tends to approach the dry-bulb temperature of the air. In this stage, it is necessary to lower the temperature in the dryer or scorching will result. Another troublemaker is case hardening. This is usually caused by too rapid drying and the outer surface becomes hard and will not let the inner moisture pass through it to be evaporated. Proper control of temperatures and humidities will overcome this.

The tunnel dehydrator, with the exception of the continuous belt type, is made up of the casing or enclosure, heating equipment, fans, food trucks, trays, and controls.

The dehydrator, itself, is of varying size, depending upon the design and capacity. It is usually constructed of insulating material so that the radiation losses will be kept to a minimum. Heating equipment in this part of the country is usually steam and blast coils. However, many tunnels are direct fired, using either gas or oil. The products of combustion pass directly over the food.

The more popular fan is the regular centrifugal type blower, which is usually needed as static pressures ordinarily run in the neighborhood of 2 in.

Trays are made in quite a few sizes and designs. The all metal tray, with the proper finish, is the best. This type has less tendency to cause sticking, is easily cleaned, and has a good heat transfer factor. Probably most of the trays in use at the present time are made of wood or wood sides, with woven wire bottoms, due to priorities. Trucks may be anything from factory fabricated to a rough skid.

Nearly every dehydrator is short on controls. Many of them have controls that are not working properly, or are not in good repair. It is not unusual to find wicks on wet-bulb thermometers caked up or the inlet valves to the pans completely clogged. I recently visited a plant where the wet-bulb wicks had not been changed for over a year. There is room for quite a bit of improvement in this phase, and it will have to come if high quality material is going to continue to be turned out.

Each dryer should have at least a wet-bulb and a dry-bulb thermometer on the entering air and a dry-bulb on the leaving air. A good many are now working satisfactorily with less, but when something goes wrong, it is pretty hard to pin down the trouble. The wet-bulb depression on the entering air will give you a definite available capacity, and the difference between the entering and leaving dry-bulb will give you some idea of how much work is being done. Naturally, it is better to use recording instruments if these can be secured, but indicating types will do. This will necessitate making a chart and keeping a permanent record of the different temperatures during the drying process.

There are several basic types of tunnels. These have been used with varying degrees of success around the country. The parallel flow tunnel derives its name from its method of operation. The food and air move in the same direction. The moist food enters the hot end of the tunnel and the hottest air hits it first. This results in very rapid drying during the initial stage of the process. This early drying tends to dissipate the heat and increase the relative humidity of the air. Consequently, the drying capacity of the air in the later stage is practically nil. It is for this reason that a complete drying cycle in the parallel flow tunnel usually takes quite a long time.

The counter flow tunnel is just the reverse as to air and food flow. In this tunnel the hottest air comes in contact with the driest food. This gives good drying in the later stages. However, the temperature of the entering air must be below the critical temperature of the product. By the time the air is passed up to the wet end of the tunnel, the capacity for work has been greatly reduced. For this reason, the drying times in the counter flow tunnel are also rather long.

Since, for the war effort, Canada has been dehydrating longer than we

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ing that product, the more efficient will be our drying operation.

Some of the heat of the air is lost in evaporating the water. At the same time the air is giving up its heat it is losing its ability to hold moisture, so obviously the air must not be cooled below the saturation temperature for the water it is carrying, nor, equally obvious, can more water be evaporated from the spray than the quantity which will correspond to the maximum vapor pressure of water at the temperature in the drying chamber. It follows that the maximum temperature of the air in the drying chamber is the temperature at which the air becomes saturated if drying is to take place.

Since the rate of evaporation decreases as we near the saturation point we are limited in the degree of saturation by the length of time the droplet and powder particle remain in the system. Since spray drying is a very rapid process we are often limited to only a fractional part of what would be saturation. The lower our moisture content requirement the lower will be this percentage of saturation. For instance, there is a stage at which an equilibrium will be established. Roughly the corresponding relative humidity for a 4 per cent moisture product is 30 per cent. Obviously, a product is not in a spray dryer long enough to reach a true equilibrium—nevertheless such a relationship exists and we must be guided by it when determining the amount of water that can be removed in a drying chamber.

Any product, regardless of its merits, must fill a real need in the commercial field in order to promote research and development of its mechanical functions.

More and more products are being introduced on the market in powdered form as the public recognizes the saving and convenience of the use of powdered products. The development and widespread use of some of these items will be brought about by the necessity of concentrated foods to carry on this present war.

The spray system has contributed greatly in this present war effort to provide high quality dried food for our fighting men and our allies, and by the same token the present expansion of dried food use has brought about changes in it to improve quality and capacity.

One particular design comprises a complete unit in that no vacuum pan is required to increase the solids content of those products which are low in solids. It is a recognized fact that the cost of making a pound of powder goes up rapidly when a liquid low in solids is sprayed into a drying chamber. The reason for this is obvious, as the cost is in direct proportion to the number of pounds of water removed, so that a product containing 10 lb of solids to each 100 lb of water would cost approximately twice as much to dry as a product containing 20 lb of solids per 100 lb of water, when the cost is based on the pound of powder basis. Therefore, the higher the concentration consistent with a product that can be atomized properly the lower the drying cost per pound of powder.

All types of dryers have a top limit to which they can go to in percentage of saturation for each particular moisture content powder. The lower the moisture requirement for the powder, the lower will be the percentage of saturation for that particular drying chamber. If the air moves counter-current to the liquid being sprayed, so that the air as it is passing to the exhaust meets the product being sprayed just before leaving the chamber, a higher humidity in the air being exhausted can be attained. This would be quite a factor in the efficiency of a drying chamber.

The spray nozzle is placed in the center of the drying chamber at the top of the cone. As the product is sprayed it comes in contact with the most humid air, because this is the air that is ready to leave through the pipe just above the nozzle. The small droplets are now moving away from the nozzle and are traveling in a cyclonic direction and at the same time they are traveling down and also toward the outside wall due to the centrifugal force. The droplet then is passing first through a humid zone and as it travels it is passing through various zones of air, each one slightly dryer until it has almost reached the cone shaped wall, when it is in a medium of the dryest air in the system. The powder is almost dry but it has still not reached the bottom of the cone. To get to the outlet at the bottom of the cone it is passing through fairly dry air of a temperature of about 180 F in the case of milk and about 145 F in the case of eggs. An automatic recording thermometer registers the temperature in the cone and it is this temperature which is used to control moisture to a very definite point. The cone temperature is the controlling factor of moisture since this is the zone where our final drying takes place. One of our principal customers at one time asked us to place the thermometers at the top of the drying chamber to record the outgoing air temperature which is usually about 160 in the case of milk. This was soon discontinued as we learned that we had precision control of our moisture when the cone temperature is controlled, but this did not hold true of a well controlled exhaust temperature.

Reviewing briefly then the path of any given droplet through the dough stage to the dry powder stage is as follows: First a very humid air just before being exhausted at a temperature of about 160 F, then through zones of dryer and hotter air, and finally through the finishing drying zone of 180 F. The particle of powder or solids never reaches the temperature of the air due to the cooling effect of the rapid evaporation. Powder leaving the drying chamber averages 50 to 60 F below the cone temperature.

As previously mentioned the exhaust air from the drying chamber is only partially saturated, and as this air leaves the drying chamber it enters the liquid collector of the system. Here the air is washed free of any fine powder particles which may come over with it, and also this air picks up additional moisture. It is not difficult to incorporate this additional moisture into the air, as the air passes through the spray of hot milk. Milk at a temperature of 170 F is sprayed into the liquid collector through small holes. These holes are large enough to give sufficient strength to the spray to carry it through the high velocity air and to the side wall where it flows back into the tank of the liquid collector from where the concentrated milk passes to the high pressure pump to the atomizing nozzle in the drying chamber.

The spray nozzle is placed just at the top of the cone and in the center of the circle. The finely atomized spray gives up its moisture very rapidly and it travels out with the air. The powder moves downward and out of the drying chamber and into the powder hopper and through the sifter and here it enters the barrel or package.

The liquid collector with its sprays of fluid milk reduces the volume of the air as it enters. Due to the incorporation of the moisture in the air and the drop in the air temperature due to the high rate of evaporation in the collector there is a material decrease in static pressure in the tunnel that connects the drying chamber and the liquid collector. Ordinarily the supposition would be that the sprays in the collector have a retarding effect on the flow of the air through it, but the exact opposite is the case.

We have always used the single spray nozzle in our drying chamber so that the shape of the spray conforms to the shape of the drying chamber, so that as each particle travels away from the nozzle radially and also in a cyclonic direction, it travels the same distance as every other particle and it does this in a very definite period of time. The high cyclonic movement of the air in the drying chamber tends to throw the powder out of the air as well as increase the distance traveled by any given particle. The fact that the cyclone throws the powder out gives a powder and air separator right in the drying chamber, thereby eliminating the necessity of doing this in a second powder hopper. As the powder leaves the drying chamber only enough air is blown into the powder to convey the powder. Part of this air is room air and a small amount of it comes from the bottom of the drying chamber with the powder.

ILLINOIS CHAPTER MEMBERS AID RATIONING BOARDS

The president of the Illinois Chapter of the ASHVE, C. E. Price, Manager, Keeney Publishing Co., Chicago, has re-appointed John Howatt, Chief Engineer, Chicago Board of Education, as chairman of the chapter's engineering advisory committee for fuel conservation for the 1943-44 season, and has offered the engineering and technical services of the chapter to the local offices of the OPA, PAW, WPB and OCD. The other members of the committee are A. B. Martin, Branch Manager, Kewanee Boiler Co., and C. M. Burnam, Jr., Editor, Heating, Piping & Air Conditioning, both of Chicago.

This committee was initiated last year by E. M. Mittendorff, Engineer, Sarco Co., Inc., Chicago, then president of the Illinois Chapter. At that time, a chapter member or another engineer was named to serve each of the war price and rationing boards in the Chicago area as technical adviser on oil rationing and conservation problems. Many of these advisers have become members of the local boards they serve as technical advisers.

Under the guidance of the chapter president and the engineering advisory committee for fuel conservation, chapter members have worked and are working in close cooperation with local boards on conversion problems requiring engineering knowledge, as well as on routine processing and tailoring of oil rations, issuance of auxiliary and hardship oil rations, and other rationing procedures. Through a series of mailings sent to chapter members engaged in this activity, Chairman Howatt has kept them informed of amendments to OPA fuel oil regulations and methods of handling particular cases. This work is to continue during the coming year.

THE ARMY NEEDS ENGINEERS' HELP

The National Inventors Council has secured the consent of the Army to the release of a list of some of the problems in which the Army is interested. A practical and effective solution of any of them will be a real contribution to the war effort. Actual war experience has shown the need of various devices or methods not practically available, and has also shown the need or the desirability of substitutes or alternatives for devices and methods which are already in our possession.

While information of the needs of the Army has only been given to agencies officially authorized to receive and use it, such a list has not heretofore been released since it might give information of what we possess or do not possess. The present list is not of this character and is being released to appropriate professional societies and research organizations in order that it may be passed along to their membership. Additional lists may be released in the future.

Suggestions on any of these problems should be submitted to the National Inventors Council, Room 1313, Commerce Building, Washington, D. C., and should comprise a clear description of the proposal with such sketches or drawings as may be necessary.

Some Problems in Which the Army Is Interested

1. A method of removing the tetraethyl lead from leaded gasoline to make it usable in stoves, lanterns and small engines.
2. An inexpensive metal suitable for Quartermaster tableware; one having requisite strength, freedom from corrosion by food acids or alkalis; durability and attractiveness.
3. Suitable substitute for rubber for insulating wire; should be flexible and durable.
4. Detectors of enemy personnel who may be approaching (unseen) on jungle trails or fences or similar barriers.
5. Sonic or supersonic means or methods of signalling in the field.
6. Improved means or methods of signalling the identification of ground troops to friendly airplanes and vice versa.
7. Improved traction devices for wheeled vehicles of all types. **Note:** Present chains and other devices are cumbersome, inconvenient to apply and remove and lack sufficient traction.
8. Tracks for tractors and other motorized equipment which will operate efficiently in snow and extreme cold.
9. Better air cleaners for use on tank engines and the like; more effective than present cleaners and requiring less maintenance.
10. Methods of quick-action waterproofing for enabling vehicles to ford water several feet deep, without stalling engines.
11. Storage battery not adversely affected by very low temperatures.
12. The detector and method of locating non-metallic land mines.
13. Equipment or methods for removing land mines rapidly from mine fields without injury to equipment or personnel.
14. Methods of rust-proofing ferrous metals, which are more durable than present methods, such as bonderizing, etc.
15. Absorbents for carbon monoxide or catalysts or other means for oxidation of this gas to render it non-injurious to personnel.
16. Means of defeating darkness to permit vision at night without aid of visible reflected light. **Note:** Probably involves an apparatus to translate infra red rays to visible light.
17. Means of long distance communication outside the present scope of radio and not restricted by line-of-sight projection.
18. Searchlights which may afford ready means for spreading the beam from narrow high intensity to 15 deg of greatest intensity practicable.
19. A simple non-toxic process for darkening aluminum and other metals; to make them non-reflectant to light.
20. Methods of sabotage by friendly inhabitants within occupied areas.

H. W. SKINNER DIES IN FORT WORTH

Word has reached us recently of the death of Henry W. Skinner, on March 24, 1943 in the Methodist Hospital. A consulting engineer of Fort Worth, Tex., he was a member of the ASHVE since 1920. Mr. Skinner was born on February 10, 1877 in Osage, Ia., where he received his early education. He was graduated from the Iowa State College, Ames, with a degree in mechanical engineering.

In 1908 he was in charge of design and specifications for distribution mains, heating plans, and construction at the U. S. Naval Training Station, Great Lakes, Ill. Completing his job there in 1911 he became superintendent in charge of the Southern Pacific Office Building in Houston, Tex., where he remained for three years. In 1912 he joined Marshall and Fox, Chicago, and as designing engineer had charge of the mechanical equipment in the C. B. & Q. office building, the Lytton and Gibbons buildings, apartment buildings on the north shore, the Home Office Building for the NW Mutual Life Insurance Co. in Milwaukee, and the Kaiserhof and Morrison hotels in Chicago.

The following year as representative for Jarvis Hunt, Chicago, he was in charge of the construction of the Dallas (Texas) Union Station, as well as the mechanical work in Chicago. He returned to the U. S. Naval Training Station in 1917, where he was in charge of the water purification and pumping works. From 1919-1922 he was in charge of all mechanical work in the Jarvis Hunt office. Having specialized in the mechanical equipment of buildings he later became associated with Rapp and Rapp, Chicago, in charge of specifications. Following this connection he became associated with Wyatt C. Hedrick, Inc., Fort Worth, where he remained for five years as mechanical engineer on over 100 projects. In 1931 he went into business for himself as consulting engineer, which he carried on until 1941 when he became mechanical engineer for the Austin Co., Fort Worth.

In a memorial to Mr. Skinner, one of the Society's North Texas Chapter members, R. K. Werner, delivered the following eulogy:

"Henry Skinner had almost attained to the Psalmist's three score and ten when he received the final summons from that world which is the bourne of all the children of earth. He led a life somewhat secluded from the public, and his qualities of mind and his fine attributes of heart were not worn as a badge, and were not conspicuous except to the few intimates who delighted in his friendship. Henry Skinner was of that type of men who frequently hide their finer sensibilities from the view of their fellowmen, in consequence of which they are so often misjudged by those who do not know them, and who are therefore in no position to judge of what lies within. You had to know him in order to appreciate and like him. And in the language of the poet Longfellow, who said,

> 'The lives of great men all remind us
> That we may make our lives sublime.
> And departing leave behind us
> Footprints on the sands of time.'

"Henry Skinner has left footprints on the sands of time.

"He had led a full life not without anxiety and care, and he had sought

to meet the vicissitudes of life's struggle bravely and with a high resolve to keep his name clean and blameless and he had no fear of the world's opinion, and could look his conscience in the face without dread and without reproach. He was in many respects as simple and as trusting as a child, in all things an honest gentleman, who sought the ways of truth and too, concealed beneath the brusque exterior there lay a mine of love and affection for his friends."

In the death of Henry W. Skinner his fellow members lost a true friend, a fine, kindly gentleman, who will be missed and mourned by all with whom he was associated.

The Officers and Council of the Society extend their sincere and heartfelt sympathy to his widow and family who survive.

DR. H. B. MELLER, AIR POLLUTION SPECIALIST, DIES

Dr. H. B. Meller, a world authority on air pollution and smoke prevention, died at the Magee Hospital, Pittsburgh, on June 27, after a long illness, at the age of 65.

Dr. Meller served as a member of the ASHVE Technical Advisory Committee on Atmospheric Impurities and Resulting Safety and Health Requirements in 1936-37, and as chairman of the sub-committee on Air Pollution and Purification in 1939, and as a member of the Air Pollution Committee in 1940.

In connection with his work at the Mellon Institute for two decades, Dr. Meller became known internationally as a specialist on air pollution, and through his efforts, Pittsburgh adopted an anti-smoke ordinance which became a model for other cities throughout the country. He was ever ready to lend his valuable assistance to the many municipal officials seeking data on smoke abatement.

From 1920 to 1938 he was in charge of Pittsburgh's Bureau of Smoke Regulation in the City's Health Department. During the depression years, in this capacity, he directed a WPA project for scientifically trained personnel, and collected a wealth of data on the city's air pollution problems.

He will be greatly missed by his many friends and associates with whom he served on the various Research Technical Advisory Committees of the Society.

SUMMARY OF LOCAL CHAPTER MEETINGS

Chapter	Meeting Date	Subject	Speakers	Other Features	Attendance	Attendance Ratio*
Southern California	June 9	Aircraft Engine Superchargers.	W. H. Allen, Engr. General Electric Co.	Installation of New Officers for 1943-44. Meetings Adjourned until Sept. 8, 1943.	38	0.30
Cincinnati	May 27	Annual Election—New Officers: Pres. Albert Buenger; Vice-Pres. G. V. Sutfin; Sec'y-Treas. E. J. Richard. Board of Governors: A. W. Edwards.	A. L. Hard.		32	0.66
Minnesota	April 5	Noon Business Meeting.		Nomination of New Officers	14	0.14
		Evening Meeting—Versatility of Plastics	Dr. C. A. Mann		26	0.27

*The attendance ratios shown represent the meeting attendance divided by the Chapter membership. These will be useful as a partial indication of interest shown by local chapter members in various types of subjects programmed by other local chapters, and may be helpful in deciding on subjects for other chapter meetings.

Candidates for Membership

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 23 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by August 15, 1943, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
Addams, Paul K., Executive Vice-President and Treasurer, Fitzgibbons Boiler Co., Inc., New York, New York.	Homer Addams W. H. Carrier	W. H. Driscoll G. E. Olsen
Almert, Harold, Consulting Eng., New York, New York.	Homer Addams James A. Donnelly	G. E. Olsen Dudley F. Parker

BERMEL, A. H., Chief Mechanical Estimating Engineer, August Arace & Sons, Inc., Elizabeth, N. J. *(Advancement)*

COULTER, THOS. H., Secretary, Vermiculite Research Institute, Chicago, Ill.

COYNE, JOHN H., Foreman, Wright Aeronautical Corp., Lockland, O.

DICKINSON, NEVILLE S., Engr. & Owner, Motor Sales & Engineering Co., Inc., Newark, N. J.

ELSIE, ROBERT L., Purchasing Agent, Atlas Engineering and Machine Co., Ltd., Toronto, Ont., Canada.

FARRELL, EDWARD J., Partner, Farrell & White, Detroit, Mich.

GILMORE, JOHN L., Owner, John L. Gilmore, Htg., Vtg., Brunswick, Ga. *(Advancement)*

HALT, HOWARD R., Office Mgr., Thos. J. Sheehan Co., St. Louis, Mo.

HAPPERFIELD, GILBERT J., Dir., Chandos Engineering Co., Ltd., London W 1, England.

HARRINGTON, D. W., Sales Engineer, Dallas Air Conditioning Co., Dallas, Tex.

LONG, D. A., Sales Engineer, Trane Co. of Canada, Ltd., Toronto, Ont., Canada.

MARTIN, WILLIAM T., Chief Engineer, Beech-Nut Packing Co., Canajoharie, N. Y.

NEILANS, J. L., Draftsman, Trane Co. of Canada, Ltd., Toronto, Ont., Canada.

POLAD, THOMAS H., Asst. Chief Engineer, Rudy Furnace Co., Dowagiac, Mich.

PRICE, WILKES C., Engineer, Six Associates, Inc., Nashville, Tenn.

ROSEBROUGH, J. STODDARD, Sales Engineer, L. J. Mueller Furnace Co., St. Louis, Mo. *(Advancement)*

SCHWARZ, GEO. C., General Manager, Thos. J. Sheehan Co., St. Louis, Mo.

SCOTT, CLARENCE E., Mgr., Industrial Heating and Refrigeration Division, Fedders Mfg. Co., Inc., Buffalo, N. Y. *(Reinstatement and Advancement)*

THOMAS, FRANK M., District Sales Manager, Oklahoma Natural Gas Co., Tulsa, Okla.

TUTTLE, ALBERT J., Draughtsman, R.A.A.F., Victoria, Australia.

WATTS, LEONARD C., Partner, J. Roger Preston & Partners, London, Eng.

L. B. Ray	*G. G. Eckstein
J. H. Broome	*A. T. Petroff
C. M. Burnam, Jr.	F. P. Keeney
C. E. Price	J. S. Kearney
Harry B. Matzen	F. S. Abbott
*Charles G. Mingley	E. J. Richard
Ernest Szekely	L. B. Ray
W. C. Kruse, Jr.	H. P. Morehouse
G. E. Cole	S. A. Jennings
H. H. Angus	V. J. Jenkinson
W. H. Old	R. S. M. Wilde
E. F. Hyde	M. B. Shea
W. J. McKinney	*W. F. Peirce
C. L. Templin	*F. N. Buzzard
H. C. Sharp	C. R. Davis
G. B. Rodenheiser	W. J. Oonk
W. G. Case	*R. G. Crittall
*T. E. Barnes	*J. R. Kell
M. L. Brown	J. A. Ray
E. T. Gessell	E. J. Stern
J. W. O'Neill	V. J. Jenkinson
S. A. Jennings	A. J. Strain
*J. S. Ellithorp	*W. F. Lampe
A. E. Stacey, Jr.	*E. W. Shineman
J. W. O'Neill	V. J. Jenkinson
S. A. Jennings	A. J. Strain
T. W. Torr	G. D. Woodhouse
J. S. Cunningham	J. C. Harden
E. K. Campbell	*W. W. Droget
Roger P. Campbell	*A. Lord
L. W. Moon	B. C. Simons
E. E. Carlson	C. E. Hartwein
H. C. Sharp	E. E. Carlson
C. R. Davis	W. J. Oonk
M. F. Blankin	*A. E. Reiss
S. A. Whitt	*Joseph Askin
F. A. Grossman	*H. C. Pierce
C. H. Dean	*J. H. Wardell
A. E. Atherton	S. R. Bell
T. A. Roseby	*J. M. Picot
W. W. Nobbs	G. R. Jackson
G. N. Haden	*R. G. Crittall

*Non-member.

Candidates Elected

In the past issues of the Journal of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

CALLAHAN, THOMAS H., Owner, Callahan Equipment Co., Inc., White Plains, N. Y. *(Reinstatement)*

DOUGLASS, THOMAS C., Mgr. & Secretary, Mech. Dept., Pacific Electrical Co., Inc., San Francisco, Calif. *(Reinstatement)*

FOOTE, AVERY G., Chief, Heat Transfer Section, Curtiss-Wright Corp., Research Laboratory, Buffalo (5), N. Y. *(Reinstatement)*

GUILLORY, J. M., Manager, Industrial & Commercial Engineering Division, New Orleans Public Service, Inc., New Orleans, La.

HODEAUX, WALTER L., Mechanical Consultant, Florida Associates, West Palm Beach, Florida. *(Reinstatement)*

HOWE, WILLIS W., Sausalito, Calif. *(Reinstatement)*

MARC, HENRI M., Asst. Dir. of Research, The Philip Carey Mfg. Co., Lockland, Cincinnati, Ohio.

MCMICHAEL, CHESTER L., Sales Engineer, H. H. Wright Co., Kansas City, Mo.

MUSSER, NEIL B., Senior Naval Architect, U. S. Maritime Commission, Oakland, Calif.

PASTEUR, HUGH W., Export Manager, J. E. Hall, Ltd., Dartford, Kent, England.

REILLY, J. HARRY, Design Engineer, Federal Shipbuilding & D. D. Co., Newark, N. J. *(Reinstatement)*

ROBINSON, K. E., Engineer, Lansing, Mich. *(Advancement)*

SCOTT, WIRT S., Air Conditioning Engr., Philadelphia Electric Co., Philadelphia, Pa.

SINISH, WILLIAM R., Design Engr., Wyatt C. Hedrick, Architect & Engineers, Fort Worth, Tex.

WALSH, JAMES A., President, Air Conditioning Co., Houston, Tex. *(Advancement)*

ASSOCIATES

BRINDELL, CARL E., Sales Representative, Cressy-Legget & Brindell, New Orleans, La.

CRESSY, L. RENE, Sales Representative, Cressy-Legett & Brindell, New Orleans, La.

GLASS, ROBERT, Secy.-Treas.-Supt., Partridge-Halliday, Ltd., Winnipeg, Man., Canada.

SCHROEDER, DELMAN E., Test Engr., Murray Iron Works Co., Burlington, Ia.

WILLIAMS, E. BRYAN, Laboratory Mgr., The Torrington Mfg. Co., Torrington, Conn.

Officers

President.....M. F. BLANKIN
First Vice-President.....S. H. DOWNS
Second Vice-President.....C.-E. A. WINSLOW
Treasurer.....E. K. CAMPBELL
Secretary.....A. V. HUTCHINSON
Technical Secretary.....CARL H. FLINK

Council

M. F. BLANKIN, *Chairman* S. H. DOWNS, *Vice-Chairman*

Three Years: J. F. COLLINS, JR., JAMES HOLT, E. N. MCDONNELL, T. H. URDAHL.

Two Years: L. G. MILLER, A. J. OFFNER, A. E. STACEY, JR., B. M. WOODS.

One Year: E. O. EASTWOOD, A. P. KRATZ, W. A. RUSSELL, L. P. SAUNDERS, C. TASKER.

Council Committees

Executive—E. O. Eastwood, *Chairman;* A. P. Kratz, E. N. McDonnell.

Finance—J. F. Collins, Jr., *Chairman;* C.-E. A. Winslow, B. M. Woods.

Membership—E. K. Campbell, *Chairman;* A. J. Offner, W. A. Russell.

Meetings—S. H. Downs, *Chairman;* James Holt, C. Tasker.

Standards—L. P. Saunders, *Chairman;* L. G. Miller, T. H. Urdahl.

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Advisory Council

E. O. Eastwood, *Chairman;* Homer Addams, D. S. Boyden, W. H. Carrier, S. E. Dibble, W. H. Driscoll, W. L. Fleisher, H. P. Gant, F. E. Giesecke, E. Holt Gurney, L. A. Harding, H. M. Hart, C. V. Haynes, E. Vernon Hill, John Howatt, W. T. Jones, D. D. Kimball, G. L. Larson, S. R. Lewis, Thornton Lewis, J. F. McIntire, F. B. Rowley and A. C. Willard.

Special Committees

Admission and Advancement: E. P. Heckel, *Chairman (one year);* T. T. Tucker *(two years);* H. Berkley Hedges *(three years).*

ASHVE-ASTM-ASRE-NRC Committee on Thermal Conductivity—F. C. Houghten, *Chairman;* C. B. Bradley, H. C. Dickinson, R. H. Heilman, E. R. Queer, F. B. Rowley, T. S. Taylor, G. B. Wilkes.

ASHVE-IES Joint Committee on Lighting and Air Conditioning—H. M. Sharp, *Chairman;* W. R. Beach, B. C. Candee, W. G. Darley, C. L. Kribs, Jr., P. M. Rutherford, Jr.

Chapter Relations—J. F. Collins, Jr., *Chairman;* L. P. Saunders, H. E. Sproull.

Constitution and By-Laws—L. T. Avery, *Chairman;* M. W. Bishop, R. A. Miller.

F. Paul Anderson Award—S. H. Downs, *Chairman;* Tom Brown, F. E. Giesecke, L. L. Lewis and F. B. Rowley.

Guide Publication—P. D. Close, *Chairman;* Thomas Chester, John James, S. Konzo, C. S. Leopold.

Publication—C. H. B. Hotchkiss, *Chairman (one year);* J. H. Walker *(two years);* L. E. Seeley *(three years).*

War Service—John Howatt, *Chairman;* W. H. Driscoll, E. N. McDonnell, L. E. Seeley, B. M. Woods.

Nominating Committee

Chapter	*Representative*	*Alternate*
Atlanta	T. T. Tucker	L. F. Kent
Cincinnati	M. E. Mathewson	E. W. McNamee
Connecticut	L. E. Seeley	H. E. Adams
Delta	G. E. May	G. C. Kerr
Golden Gate	B. M. Woods	
Illinois	E. M. Mittendorff	A. O. May
Iowa	F. E. Triggs	C. W. Helstrom
Kansas City	L. T. Mart	M. M. Rivard
Manitoba	Ivan McDonald	Einar Anderson
Massachusetts	E. G. Carrier	A. C. Bartlett
Michigan	M. B. Shea	S. S. Sanford
Minnesota	R. E. Backstrom	William McNamara
Montreal	G. P. Ste-Marie	R. R. Noyes
New York	H. H. Bond	C. S. Koehler
Nebraska	B. G. Peterson	W. R. White
North Carolina	F. E. P. Klages	Arvin Page
North Texas	T. H. Anspacher	
Oklahoma	E. T. P. Ellingson	E. F. Dawson
Ontario	V. J. Jenkinson	H. R. Roth
Oregon	B. W. Farnes	
Pacific Northwest	R. D. Morse	E. H. Langdon
Philadelphia	H. H. Mather	Edwin Elliot
Pittsburgh	E. C. Smyers	E. H. Riesmeyer, Jr.
St. Louis	C. F. Boester	M. F. Carlock
South Texas	J. A. Walsh	A. J. Rummel
Southern California	W. O. Stewart	Leo Hungerford
Washington, D. C.	J. W. Markert	F. A. Leser
Western Michigan	T. D. Stafford	L. G. Miller
Western New York	H. C. Schafer	S. M. Quackenbush
Wisconsin	C. H. Randolph	Ernest Szekely

Officers of Local Chapters

Atlanta: Organized, 1937. Headquarters, Atlanta, Ga. Meets, First Monday. President, A. H. Koch, 3687 Peachtree Rd. Secretary, L. F. Lawrence, Jr., 304-101 Marietta St., Atlanta 3. **Cincinnati:** Organized, 1932. Headquarters, Cincinnati, O. Meets, Second Tuesday. Honorary President, Capt. C. E. Hust, President, Albert Buenger, Hotel Gibson. Secretary, E. J. Richard, 2137 Reading Rd., Cincinnati 2. **Connecticut:** Organized, 1940. Headquarters, New Haven, Conn. President, C. J. Lyons, Wilson Ave., S. Norwalk. Secretary, J. R. Smak, 160 Morehouse Highway, Fairfield. **Delta:** Organized, 1939. Headquarters, New Orleans, La. Meets, Second Tuesday. President, L. V. Cressy, 916 Union St., New Orleans 13. Secretary, J. S. Burke, 317 Baronne St., New Orleans 9. **Golden Gate:** Organized, 1937. Headquarters, San Francisco, Calif. Meets, First Wednesday. President, R. A. Folsom, 150 Hooper St. Secretary, R. B. Holland, 1275 Folsom St. **Illinois:** Organized, 1906. Headquarters, Chicago, Ill. Meets, Second Monday. President, Chas. E. Price, Room 1605, 6 N. Michigan Ave., Chicago 2. Secretary, C. M. Burnam, Jr., Room 1605, 6 N. Michigan Ave., Chicago 2.

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Iowa: Organized, 1940. Headquarters, Des Moines, Ia. Meets, Second Tuesday. President, C. W. Helstrom, 1614 Thompson. Secretary, B. E. Landes, 1603 47th St., Des Moines 10. **Kansas City:** Organized, 1917. Headquarters, Kansas City, Mo. Meets, Second Monday. President, M. M. Rivard, 4550 Main St. Secretary, D. M. Allen, 310 Board of Trade Bldg., Kansas City 6. **Manitoba:** Organized, 1935. Headquarters, Winnipeg, Man. Meets, Third Thursday. President, P. L. Charles, 206 Tribune Bldg. Secretary, Einar Anderson, 152 Bannerman Ave. **Massachusetts:** Organized, 1912. Headquarters, Boston, Mass. Meets, Third Tuesday. President, E. G. Carrier, 704 Statler Bldg. Secretary, R. T. Kern, 51 Claflin St., Leominster. **Michigan:** Organized, 1916. Headquarters, Detroit, Mich. Meets, First Monday after 10th of Month. President, S. S. Sanford, 2000 Second Ave. Secretary, A. E. Knibb, 1003 Maryland Ave., Detroit 30. **Minnesota:** Organized, 1918. Headquarters, Minneapolis, Minn. Meets, First Monday. President, William McNamara, 850 Cromwell Ave., St. Paul. Secretary, A. W. Schultz, 240 Seventh Ave., S., Minneapolis.

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Montreal: Organized 1936. Headquarters, Montreal, Que. Meets Third Monday. President, F. A. Hamlet, 1010 St. Catherine St., W. Secretary, R. R. Noyes, 630 Dorchester St., W. **Nebraska:** Organized, 1940. Headquarters, Omaha. Meets, Second Tuesday. President, G. E. Merwin, 5012 Parker St. Secretary E. F. Adams, 1227 So. 52nd St. **New York:** Organized, 1911. Headquarters, New York, N. Y. Meets, Third Monday. President, R. A. Wasson, 500 Fifth Ave. Secretary, P. G. Griess, 189 Walnut Ave., Bogota, N. J. **North Carolina:** Organized, 1939. Headquarters, Durham, N. C. Meets, Quarterly. President, F. J. Reed, 263 College Station. Secretary, C. Z. Adams, 312 Piedmont Bldg., Greensboro. **North Texas:** Organized, 1938. Headquarters, Dallas, Tex. Meets, Second Monday. President, M. L. Brown, 3500 Commerce St. Secretary, E. T. Gessell, Thomas Bldg. **Northern Ohio:** Organized, 1916. Headquarters, Cleveland, O. Meets, Second Monday. President, P. D. Gayman, 2142 E. 19th St. Secretary, G. B. Priester, Case School of Applied Science, Cleveland 6.

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Oklahoma: Organized, 1935. Headquarters, Oklahoma City, Okla. Meets, Second Monday. President, E. F. Dawson, University of Oklahoma, Norman. Secretary, E. T. P. Ellingson, 314 Savings Bldg., Oklahoma City 2. **Ontario:** Organized, 1922. Headquarters, Toronto, Ont. Meets, First Monday. President, W. C. Kelly, 602 King St., W. Secretary, H. R. Roth, 57 Bloor St., W. **Oregon:** Organized, 1939. Headquarters, Portland, Ore. Meets, Thursday after First Tuesday. President, J. A. Freeman, 1623 S. E. 11th Ave. Secretary, G. H. Risley, 516 S. W. Oak St., Portland 4. **Pacific Northwest:** Organized, 1928. Headquarters, Seattle, Wash. Meets, Second Tuesday. President, R. D. Morse, 4404 White Bldg. Secretary, J. D. Sparks, 7331 W. Green Lake Way, Seattle 3. **Philadelphia:** Organized, 1916. Headquarters, Philadelphia, Pa. Meets, Second Thursday. President, Edwin Elliot, 560 N. 16th St. Secretary, E. H. Dafter, Room 2211, 12 S. 12th St., Philadelphia 7. **Pittsburgh:** Organized, 1919. Headquarters, Pittsburgh, Pa. Meets, Second Monday. President, G. G. Waters, 1841 Oliver Bldg. Secretary, E. H. Riesmeyer, Jr., 231-33 Water St., Pittsburgh 22.

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St. Louis: Organized, 1918. Headquarters, St. Louis, Mo. Meets, First Tuesday. President, C. F. Boester, 101 E. Essex, Kirkwood. Secretary, B. C. Simons, Rm. 706, 4030 Chouteau, St. Louis 10. **South Texas:** Organized, 1938. Headquarters, Houston, Texas. Meets, Third Friday. President, A. M. Chase, Jr., Box 359. Secretary, A. F. Barnes, 602 Kirby Bldg., Houston 2. **Southern California:** Organized, 1930. Headquarters, Los Angeles, Calif. Meets, Second Wednesday. President, W. O. Stewart, 153 W. Avenue 34. Secretary, Maron Kennedy, 5051 Santa Fe Ave. **Washington, D. C.:** Organized, 1935. Headquarters, Washington, D. C. Meets, Second Wednesday. President, S. L. Gregg, 4828 Edgemoor Lane, Bethesda, Md. Secretary, A. S. Gates, Jr., 111 County Rd., Kensington, Md. **Western Michigan:** Organized, 1931. Headquarters, Grand Rapids, Mich. Meets, Second Monday. President, C. H. Pesterfield, Michigan State College, East Lansing. Secretary, V. H. Hill, 2111 Colvin Court, Lansing 10. **Western New York:** Organized, 1919. Headquarters, Buffalo, N. Y. Meets, Second Monday. President, S. M. Quackenbush, 117 W. Tupper St. Secretary, Herman Seelbach, Jr., 45 Allen St. **Wisconsin:** Organized, 1922. Headquarters, Milwaukee, Wis. Meets, Third Monday. President, F. W. Goldsmith, 513 E. Day Ave. Secretary, E. W. Gifford, 611 N. Broadway.